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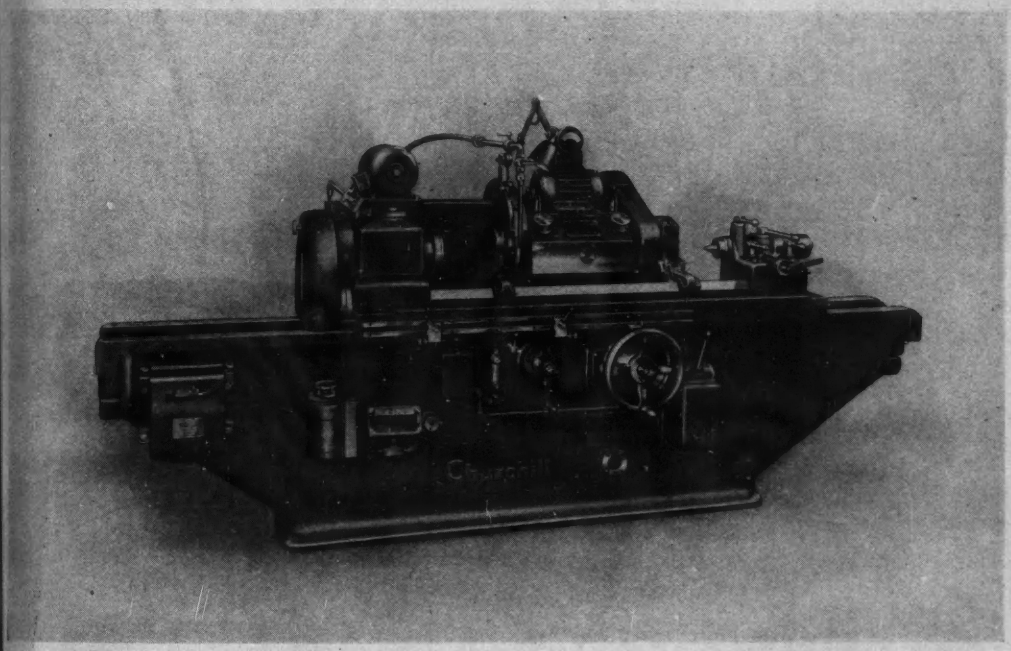

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CONTENTS

page

Alternating Premagnetization of Cores 348
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 348.

The Calibration of Venturi Meters 364
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 364.

Classified Abstracts 367
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 367.

The Dimensioning of Automobile Radiators 346
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 346.

The Future of the Czech Motor Car Industry 364
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 364.

High-Speed and Super-Speed Cutting of Metals 361
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 361.

New Methods of Melting in the Cupola 366
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 366.

A New Rigidity Formula for Crankshafts 337
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 337.

News of the Month 371
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 371.

The Normal Cathode Fall for Molybdenum and Zirconium in the Rare Gases 355
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 355.

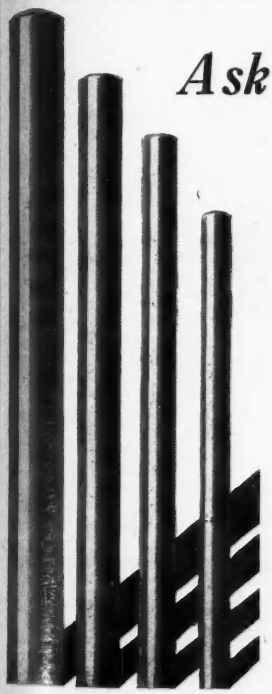
Optimum Blade Pitch for Turbo-Machines with special reference to Blades of great Curvature 358
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 358.

The Physical Mechanism of the Spark 341
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 341.

The Rotating Electric Arc Furnace 360
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 360.

Shutter-Type Dust Collector of small Dimensions 353
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 353.

Static Balancing of Turbine Wheels 351
"*The Engineers' Digest*," Vol. 7, No. 11, *p.* 351.



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A NEW RIGIDITY FORMULA FOR CRANKSHAFTS.

By S. S. ZIMANENKO. (From *Vestnik Ingenerov i Tehnikov*, No. 2, February, 1946, pp. 51-58, 8 illustrations.)

RECENT CONTROVERSIES.

THE increase in number of cylinders and speed of modern engines has resulted in ever growing resonance problems and in the introduction of various damping devices the proportions of which are first calculated and then tested experimentally. Hence the need for accurate calculation of torsional oscillations.

The simplest and the only practicable method of making these calculations is by working out the frequency of free oscillation of an equivalent compound torsional pendulum—equivalent, that is, with respect to both kinetic energy and torsional stiffness of the component parts. Thus, correct calculation of stiffness is an essential part of engine design.

Until 1933 the problem was approached entirely on the assumption that the torsion may be regarded as due to moments applied at the two ends of the crankshaft about its main axis, which we shall call Type I torsion, causing angles of twist φ. Then Prof. Grammel put forward a theory, which for a time gained ground, that calculations should be based on consideration of tangential forces applied at the crank-pins, which we shall refer to as Type II torsion, resulting in angular deformation φ_{II}.

Fig. 2 shows how the ratio $\frac{\varphi}{\varphi_{II}}$ varies with crankshafts of different dimensions, the symbols used being explained in Fig. 1, and the bottom scales being based

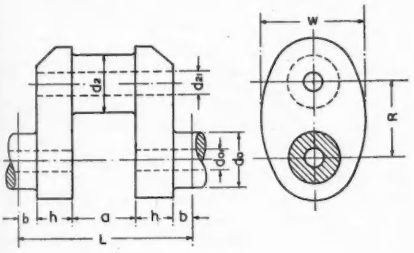

Fig. 1. Symbols used in the text.

on varying one dimension at a time in an arbitrarily chosen "average crankshaft" represented by point A, and having

$a = 2b = d_0 = d_2 = R = 1$; $W = 1{\cdot}2$ and $h = 0{\cdot}5$.

It will be noted that at point A the ratio φ/φ_{II} is 1·34 and that, in general, it is a very variable quantity, particularly susceptible to changes in d_2 and nearly always greater than unity.

In 1936 a stiffness formula based on Type II torsion was put forward but it was extraordinarily complicated, requiring about one hundred calculations for a four-cylinder crankshaft, and further calculations rising as the square of the number of cylinders. Moreover, the problem of giving a truly general theoretical representation of the physical facts was not solved as the authors adopted a number of simplifying assumptions as to the shape, working conditions and mounting of the engine.

Theories based on Type II torsion never gained much credence because experimental results always seemed to justify the older formulæ and finally a series of experiments in 1938-39 at Stuttgart have convinced Prof. Kimmel himself, who was one of the earliest exponents of Type II, that for modern high speed engines Type I calculations give better results.

Further experiments by Haug, published in 1940, indicate that the results most consistent with torsiometer indications are those obtained when shafts are given a static load of Type I or are tested dynamically. Average error is about 4 per cent. When Type II loading is applied, on the other hand, results are very inconsistent varying by up to 33 per cent.

As to the proposed Type II formulæ, they give results which vary by as much as 150-200 per cent from torsiometer observation.

SEARCH FOR A NEW FORMULA.

As a result of the above developments it was decided :

(a) To discard Type II theory.
(b) To base research on Type I loads.
(c) To carry it out by means of static loads only, the results of which were proved by Haug's investigations to be in no way inferior to dynamic tests.

Existing formulæ are usually designed to find l_0, the "equivalent length" of shaft between two adjacent main bearings. If M is the torque, φ the angle of twist and C the "stiffness" $= \frac{M}{\varphi}$, then $l_0 = \frac{1}{C}\frac{\pi}{32} G\,(d_0^4 - d_{01}^4)$ and the basis of most crankshaft stiffness formulæ is the addition of twists in the round portions plus bending in the webs, all reduced to the basis of main shaft stiffness i.e. :

$$l_0 = 2b + a\,\frac{d_0^4 - d_{01}^4}{d_2^4 - d_{21}^4} + 0{\cdot}93\,R\,\frac{d_0^4 - d_{01}^4}{hw^3} \quad .. \qquad (1)$$

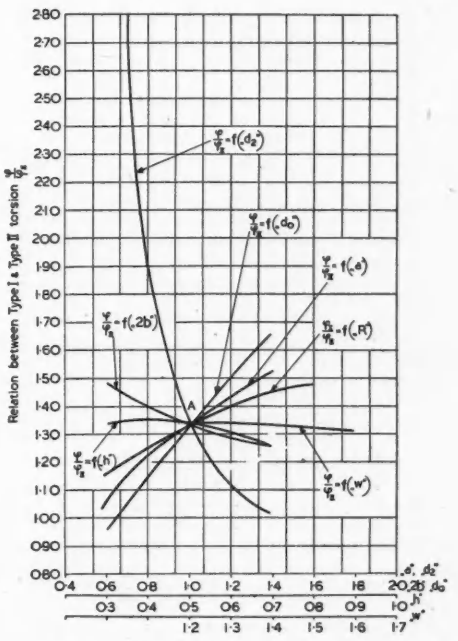

Fig. 2. Relation between Type I and Type II torsion with varying dimensions of crankshaft.

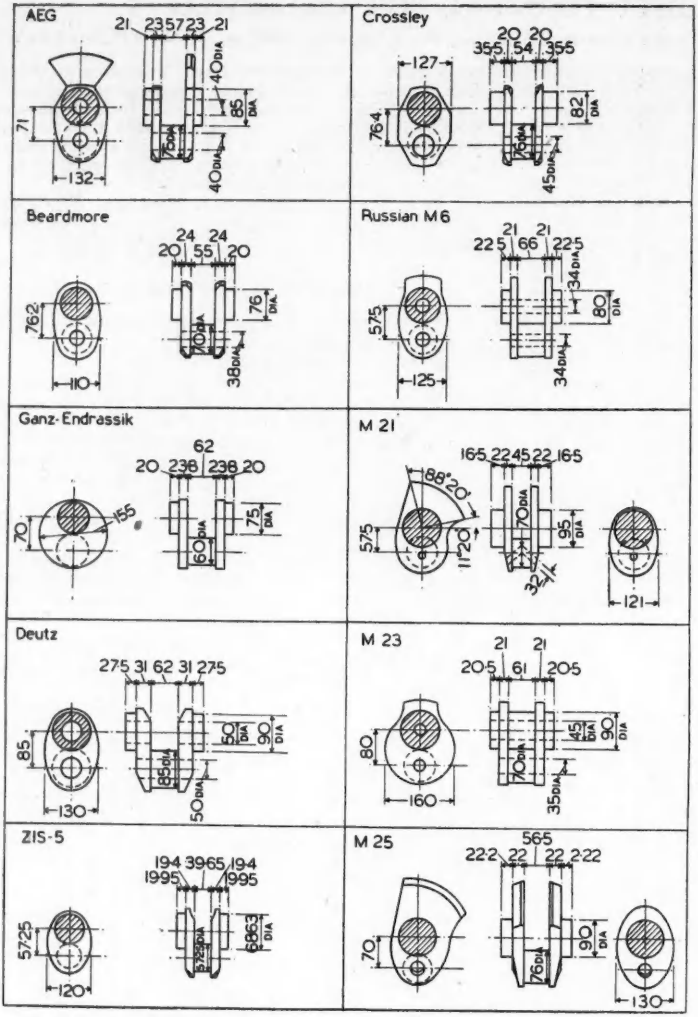

Fig. 3. **Dimensions of crankshafts tested by N.A.T.I.**

The three types of formulæ in vogue are :

(1) Three-factor formulæ as above with slight empirical modifications. Of these the best are :—

Carter's :—

$$l_0 = (2b + 0{\cdot}8h) + 0{\cdot}75a\,\frac{d_0^4 - d_{01}^4}{d_2^4 - d_{21}^4} + 1{\cdot}5\,R\,\frac{d_0^4 - d_{01}^4}{hw^3} \quad (2)$$

Timoshenko (for clearanced bearings) :—

$$l_0 = (2b + 0{\cdot}9h) + (a + 0{\cdot}9h)\,\frac{d_0^4 - d_{01}^4}{d_2^4 - d_{21}^4} + 0{\cdot}93R\,\frac{d_0^4 - d_{01}^4}{hw^3} \quad (3)$$

and Wilson's :—

$$l_0 = (2b + 0{\cdot}4d_0) + (a + 0{\cdot}4d_0)\,\frac{d_0^4 - d_{01}^4}{d_2^4 - d_{21}^4} + \\ + [R - 0{\cdot}2\,(d_0 + d_2)]\,\frac{d_0^4 - d_{01}^4}{hw^3} \quad .. \quad (4)$$

(2) Formulæ still based on equ. (1) but introducing additional items and more complicated corrections, such as, Sass, Timoshenko (for bearings without clearance), Norman and Stinson.

(3) Formulæ based on both Type I and Type II torsion of which the best is Taplin's.

(4) Purely empirical formulæ which are hardly intended as more than rough approximations and usually cover only a limited range of engines.

Of all these formulæ Carter's is about the simplest and most accurate, but the experiments on which it was based are somewhat out of date; Carter himself foresaw that his formula would either have to be modified or replaced.

One respect in which all these formulæ tend to err is that corrections for localized stresses at changes of section are made in the form of additions to the length. This is not altogether justified because recent research and the author's own experiments point to the fact that the rigidity of a transition-section is mainly dependent upon the relation between the two cross-sections and not upon the lengths of the parts.

To arrive at a new formula, experiments were conducted in the N.A.T.I. laboratory on a number of diesel crankshafts of 1933—1940 lorries of 3 to 5 ton capacity, the dimensions of which are shown in Fig. 3. At the same time dimensions of crankshafts used by Carter and other authorities were also tabulated and the results they gave were taken into consideration.

Fig. 4, shows how the various dimensions compare. Crankshaft M-21 is shown separately because it is regarded as particularly characteristic of modern trends in lorry-engine design.

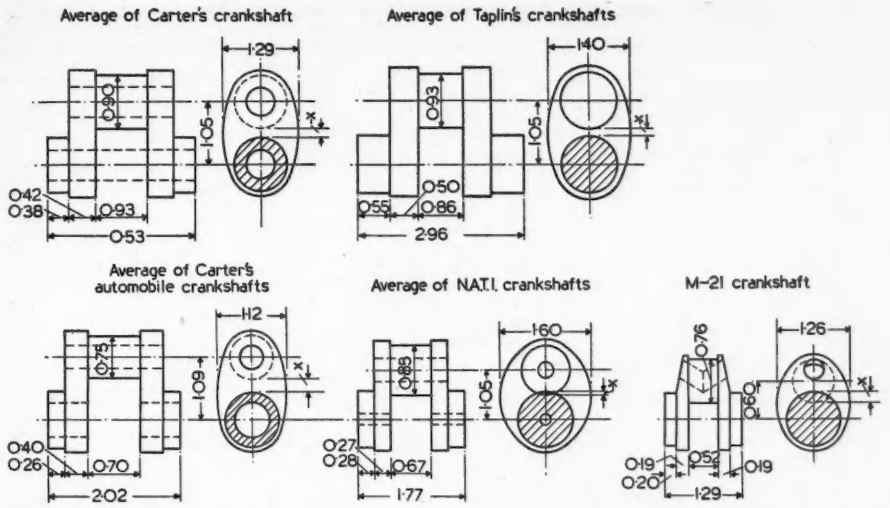

Fig. 4. **Average dimensions of N.A.T.I. crankshafts and those of earlier investigators. (All expressed in terms of $d_0=1$)**

The modern tendency to obtain stiffness by increasing diameters until the dimensions of the crank pin and the main shaft overlap, makes it necessary to introduce this overlap " x " into the calculation, or rather a " lap coefficient ", y_0,

$$\text{where} \quad y_0 = 0{\cdot}5\frac{d_0+d_2}{R} = 1+\frac{x}{R} \quad .. \quad (5)$$

It will be noted that if y_0 is greater than unity there is overlapping, otherwise x is negative, i.e. the diameters are short of overlapping by the amount x.

Table I shows x and y values for the various crankshafts under consideration and illustrates very forcibly the increasing tendency towards overlapping.

TABLE I.

Group of crankshaft	Carter's	Taplin's	Carter's Automobile Engines	N.A.T.I.	M-21
Avge value of x	—0·110	—0·065	—0·215	+0·071	+0·265
" " y	0·896	0·942	0·788	1·105	1·436

Obviously when there is overlapping there can be no pure bending in the webs and the deformation is decreased in a manner more marked than can be expected from the older formulæ, but it is difficult to assess the effect by theoretical calculations. Taking the formulæ of Carter, Taplin, Timoshenko and Wilson, and calculating l_0/d_0 for " average " crankshafts, we find that for Carter's crankshafts the various formulæ agree to within 4 per cent, with Taplin's crankshafts the discrepancy is 18 per cent, and with the N.A.T.I. average crankshaft it is as much as 29 per cent. This illustrates the failure of the older formulæ to cover modern trends.

PROPOSED NEW FORMULA.

It was felt that this should :—

(a) Be based on Type I torsion and contain the three components corresponding to twist in main shaft, crank-pin, and web.

(b) Contain allowances for all sources of weakness at changes of section.

(c) Cover adequately the " overlapping " type of crank-pin.

(d) Be reasonably simple.

(e) Be reliable and universal, i.e. not applicable to motor-vehicle shafts alone, although based mainly on them.

The search for such a formula was conducted by a series of successive approximations, first finding the best coefficients to fit the experimental results obtained with the nine N.A.T.I. crankshafts and then modifying it to fit the 15 Carter crankshafts (5 marine, 5 aviation, 5 automobile), the 4 Taplin (large diesel) and two Russian aviation crankshafts.

In all 55 tentative formulæ were employed, values calculated from them checked up against experimental results and percentage deviations tabulated. The formula finally chosen and recommended is :—

$$l_0 = \left(2b+0{\cdot}3\frac{h}{b}-d_0\right)+\left(0{\cdot}8a+0{\cdot}2\frac{w}{R}d_0\right)\frac{d_0^4-d_{01}^4}{d_2^4-d_{21}^4}+$$
$$+\frac{R}{\Phi}\frac{d_0^4-d_{01}^4}{hw^3} \quad \text{where } \Phi=\sqrt{\frac{d^2}{R}} \quad .. \quad (6)$$

Comparative results obtained by the use of this and the older formulæ are given in Table II.

It will be seen that each of the older formulæ gives an error exceeding the permissible 6 per cent in some 30-40 per cent of cases in each group, and that the proposed formula is much more accurate. Table III shows that the results it gives are better even for Carter's and Taplin's crankshafts than the figures obtained from their own formulæ.

ANALYSIS OF PROPOSED FORMULA.

The interesting features of the new formula are the subsidiary factors of the first and second main components, each expressing change of section, and the factor Φ. Otherwise the formula differs from the older formulæ only in the value of its coefficients.

(a) The factor $0{\cdot}3\frac{h}{b}-d_0$ takes the place of Carter's $0{\cdot}8h$, Wilson's $0{\cdot}4\,d_0$ etc. The difference lies in the fact that a constant coefficient (0·8 or 0·4) is replaced by a variable one, depending on the correlation of the two sections. This will be more obvious if we write

$$0{\cdot}3\frac{h}{b}-d_0 \text{ as } 0{\cdot}3\left(\frac{d_0}{b}\right)h$$

Carter's constant $0{\cdot}8h$ is very near the truth for all the older crankshafts, but for the more modern designs it would require to be modified to 1·1-1·2. This is so because h is reduced in modern engines.

TABLE II.—"EQUIVALENT LENGTH" VALUES FOUND EXPERIMENTALLY COMPARED WITH VALUES GIVEN BY VARIOUS FORMULÆ.

	Group	Crank-shaft	d_0 cm	$\frac{R}{d_0}$	$\frac{d_{31}}{d_0}$	$\frac{d_2}{d_0}$	$\frac{d_{21}}{d_0}$	$\frac{b}{d_0}$	$\frac{h}{d_0}$	$\frac{a}{d_0}$	$\frac{L}{d_0}$	Experimental $\frac{l_0}{d_0}$	Ratio of experimental l_0 to calculated value: Carter	Taplin	Wilson	Timoshenko	Proposed new formula
1	Carter	1M Marine	4·200	0·868	0·490	1·000	0·490	0·529	0·490	1·130	1·510	3·194	1·054	1·016	0·984	0·908	1·048
2		2M	23·000	0·876	0·456	1·000	0·456	0·521	0·458	1·167	1·655	3·120	1·082	1·032	0·966	0·916	1·040
3		3M	3·810	0·896	0·541	1·000	0·541	0·506	0·489	1·010	1·330	3·104	0·96	0·943	0·965	0·869	1·007
4		4M	28·500	0·931	0·461	1·000	0·461	0·507	0·457	1·126	1·648	3·050	0·059	1·015	0·957	0·906	1·030
5		5M	17·780	1·000	0·464	1·000	0·464	0·572	0·571	1·140	1·571	2·930	0·936	0·925	0·877	0·790	0·946
6		6A Aviation	7·300	0·826	0·587	0·933	0·652	0·163	0·424	0·892	1·022	4·330	1·048	0·992	1·220	1·045	1·059
7		7A	5·900	1·100	0·645	0·848	0·538	0·406	0·322	1·015	1·220	4·340	0·898	0·828	0·876	0·860	0·908
8		8A	12·700	1·200	0·650	0·850	0·500	0·300	0·320	0·660	1·190	4·175	0·932	0·990	1·005	0·975	0·950
9		9A	7·416	1·280	0·714	0·990	0·754	0·341	0·300	1·010	1·145	5·080	1·025	0·995	1·220	1·160	1·134
10		11A	13·000	1·540	0·692	1·040	0·692	0·469	0·423	1·016	1·231	3·960	0·952	0·900	1·100	1·026	1·030
11		13C Auto-mobile	6·667	1·047	0·571	0·810	0·429	0·286	0·410	0·705	1·152	4·500	1·038	1·048	1·013	0·946	1·006
12		14C	5·080	1·125	0·625	0·750	0·375	0·263	0·400	0·750	1·100	5·500	1·070	1·060	1·013	0·963	1·009
13		15C	5·405	1·058	0·588	0·706	0·353	0·235	0·424	0·635	1·094	5·700	1·108	1·058	0·961	0·910	0·992
14		16C	5·080	1·125	0·625	0·750	0·438	0·263	0·375	0·725	1·125	6·750	1·310	1·246	1·213	1·170	1·225
15		17C	4·800	1·353	0·423	0·937	0·000	0·544	0·405	2·592	1·243	5·780	0·910	0·926	0·900	0·864	0·945
16	Taplin	No. 18	19·050	1·000	0·000	1·033	0·000	0·500	0·437	0·867	1·233	2·980	0·790	0·920	0·921	0·821	0·908
17		No. 19	20·300	0·938	0·000	0·843	0·000	0·469	0·407	0·812	1·220	3·970	0·908	0·944	0·881	0·826	0·920
18		No. 20	35·500	1·142	0·000	1·000	0·000	0·643	0·661	0·858	1·680	3·090	1·028	1·088	0·970	0·841	1·040
19		No. 21	21·000	1·120	0·000	1·000	0·000	0·571	0·500	0·905	1·480	3·260	1·000	1·076	0·974	0·908	1·041
20	N.A.T.I.	AEG	8·500	0·836	0·471	0·883	0·471	0·247	0·271	0·671	1·471	3·030	1·040	0·986	0·954	0·991	0·953
21		G. Endr.	7·500	0·934	0·000	0·800	0·106	0·267	0·318	0·826	1·960	4·060	1·405	1·185	0·973	0·968	1·022
22		Deutz	9·000	0·945	0·555	0·945	0·556	0·311	0·345	0·689	1·356	2·943	0·962	0·972	0·978	0·935	0·972
23		ZIS-5	6·663	0·859	0·000	0·857	0·000	0·300	0·291	0·594	1·440	3·230	1·028	1·050	0·946	0·959	0·958
24		Crossley	8·200	0·932	0·000	0·928	0·549	0·433	0·244	0·659	1·550	3·480	1·032	0·994	0·994	1·023	1·023
25		M-6	8·500	0·894	0·400	0·824	0·450	0·271	0·235	0·789	1·460	4·020	1·048	0·975	0·934	0·979	0·961
26		M-21	9·500	0·605	0·000	0·737	0·337	0·195	0·189	0·516	1·225	5·200	1·073	1·062	1·088	1·123	1·007
27		M-23	9·000	0·889	0·500	0·800	0·438	0·230	0·222	0·684	1·724	3·620	1·215	1·019	0·924	1·041	0·958
28		M-25	9·000	0·778	0·000	0·845	0·000	0·247	0·244	0·628	1·360	3·860	1·105	1·121	1·082	1·090	1·051
29	Russian Aviation	No. 1 Z.I.A.M.	9·500	1·000	0·589	0·843	0·442	0·309	0·271	0·737	1·574	3·250	1·043	0·938	0·884	0·933	0·954
30		No. 2 Z.I.A.M.	9·000	0·944	0·780	0·756	0·489	0·317	0·200	0·849	1·667	3·990	1·228	0·841	0·916	1·048	1·009

(*b*) The factor $0{\cdot}2\frac{w}{R}d_0$ is similar to Timoshenko's $0{\cdot}9h$ and Wilson's $0{\cdot}4\,d_0$ but the use of a variable function instead of a constant coefficient again contributes towards rendering the formula more universal. It is found that $0{\cdot}2\frac{w}{R}$ changes mainly with the type of machine, averaging 0·35 for Carter's marine, 0·195 for Carter's aviation, 0·2 for Carter's automobile, 0·27 for Taplin's and 0·37 for N.A.T.I. crankshafts.

(*c*) The factors discussed above are "form factors" in some way but Φ is *the* form factor deliberately introduced by the author to express the overlap position and thus to permit of using the same formulæ for the overlapping and the non-overlapping designs which would otherwise hardly be fair.

Admittedly $\sqrt{\frac{d_2}{R}}$ is not a direct measure of overlap but for all practical purposes it differs only by fractions of a per cent from

$$0{\cdot}5\frac{d_0+d_2}{R}\sqrt{\frac{R}{d_0}}$$

which would be the direct, but rather complicated way of using:

$$y_0 = 0{\cdot}5\frac{d_0+d_2}{R}$$

The relation between Φ and y_0 is $\Phi = \sqrt[n]{y_0}$ where $n = 2{\cdot}2$ to $2{\cdot}7$.

Thus the factor Φ, like the other subsidiary factors, helps to make the formula more universal.

TABLE III.—DISCREPANCIES BETWEEN CALCULATED AND EXPERIMENTAL RESULTS

Discrepancies for groups of crankshafts.	Comparison of formulæ	Proposed new formula	Carter	Taplin	Wilson	Timoshenko
Carter's crank-shafts	Average discrepancy %	5·6	7·9	5·8	8·6	9·9
	No. of discrepancies exceeding 6%	3	8	4	7	10
	No. of discrepancies exceeding 10%	2	3	2	4	6
Taplin's crank-shafts	Average discrepancy %	6·3	8·2	7·3	6·3	15·1
	No. of discrepancies exceeding 6%	2	2	3	2	4
	No. of discrepancies exceeding 10%	0	1	0	1	3
N.A.T.I. crank-shafts	Average discrepancy %	3·3	10·9	5·7	5·2	5·0
	No. of discrepancies exceeding 6%	0	4	3	4	3
	No. of discrepancies exceeding 10%	0	3	2	0	1
Russian Aviation crank-shafts	Average discrepancy %	3·8	13·5	11·0	10·0	5·7
	No. of discrepancies exceeding 6%	1	1	2	2	1
	No. of discrepancies exceeding 10%	0	1	1	1	0
All 30 crank-shafts	Average discrepancy %	4·9	9·2	6·3	7·4	9·0
	No. of discrepancies exceeding 6%	6	15	12	15	18
	No. of discrepancies exceeding 10%	2	8	5	6	10

CONCLUSIONS.

The proposed new formula is simple, universal, and, while giving more accurate results even for the older type of crankshaft, is particularly suitable for the more modern types of engine for which even the best of the older formulæ is inadequate.

THE PHYSICAL MECHANISM OF THE SPARK.

By S. TESZNER. (From *Bulletin de la Société des Électriciens*, Vol. 6, No. 56, February, 1946, pp. 61-80, 14 illustrations.)

I. INTRODUCTION.

THE physical mechanism of the spark can conveniently be divided into two distinct phases ; the first including the formative conditions, i.e. the conditions during the forming or re-forming of the spark, and the second its " established " state. In the case of sparks resulting from a d.c. potential, or from an a.c. potential of low and medium frequency, the formative and the established states or phases differ considerably, and the mechanism of the established phase is well understood. On the other hand, if the spark is caused by an a.c. potential of a high frequency such that the duration of the transient phase of the spark is no longer negligible with regard to the duration of the half-cycle itself, the established phase of the spark assumes some of the characteristics of the formative phase ; as a result, the phenomena become very complex, and, so far, do not seem to have been thoroughly investigated. To these conditions particular reference is made in the second part of this article.

II. THE STRIKING OF THE SPARK.

(a) *General.*

The process of the striking of a spark can be divided into three quite distinct phases (Fig. 1) which will be defined as follows : Phase I begins when the minimum striking potential U_d is reached (point A), and ends when the resulting current reaches an intensity sufficient to be noticeable on the oscillograph as a drop in potential (point B), a definition which, however, is not absolutely exact, as it depends on the impedance of the circuit. This first phase is called the time lag of the spark. Phase II lasts from the end of phase I until the voltage across the gap drops to either the " normal " or the " abnormal" cathode fall potential. These potentials are of the order of 300V, and up to 2000V respectively. Normally this phase is not particularly marked (B-C in Fig. 1). Phase III (C-D), lasts from the end of phase II until the voltage across the gap has dropped to a value of the order of 30 to 50V, which is only a little higher than the normal arc voltage to which it will eventually fall.

Phases II and III which usually merge together represent the " period of formation " of the spark.

(b) *Phase* I.

Phase I can, in turn, be considered to consist of two stages ; the first from reaching the minimum striking voltage U_d until the emergence of the first electron capable of initiating an " electron avalanche ", and the second stage when the " electron avalanche " has reached such proportions that it carries a current recognizable on the oscillograph.

Various theories have been put forward as to the causes of the emergence of the initial electron ; ions present in the gap, photo-electric action, ionising rays from outside, and others. So far, however, none of these theories could be upheld. It is therefore suggested that the initial electron is extracted from the surface of the cathode by the electric field. This, however, requires a field strength on the surface of the cathode of several hundred kV/cm, which is probably attained by local field concentrations on surface crystals. This theory of " auto-emission " is supported by the fact that the nature of the surface of the cathode effects the time lag, and in particular that semi-conducting or insulating particles, e.g. carborundum on the surface, inevitably reduce the time lag.

The duration of the first part of phase I cannot be determined accurately ; approximate measurements by means of photo-cells, however, have shown that it may be well below 10^{-8} sec if a very steep voltage wave is applied.

Townsend's well-known theory of " ionisation by collision " may be applied to the second part of phase I, the formation of the " electron avalanche ", but only in broad outline, as it does not explain the very short time lags found experimentally. According to Townsend, the number of electrons increases as an exponential function :

$$n = n_0 e^{\alpha x}$$

where n_0 is the original number and n the actual number of electrons, x the distance covered by the first electron and α a coefficient which is a function of the electric field, of the kinetic energy, of the electrons, and of the kind and pressure of the gas present (Fig. 2).

This theory is inaccurate, due to the role it attributes to the positive ions ; for it says in fact, that a discharge can begin only after the positive ions produced in the gap have actually reached the cathode and have liberated fresh electrons (that means, only after a time at least equal to the mean time required by an ion to travel from the anode to the cathode).

The number of electrons liberated is given by :

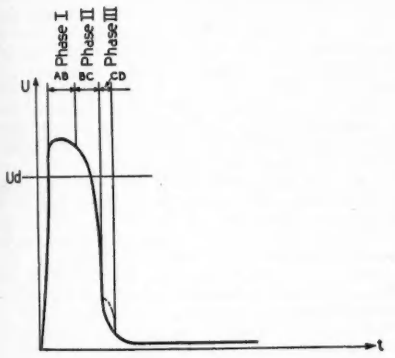

Fig. 1

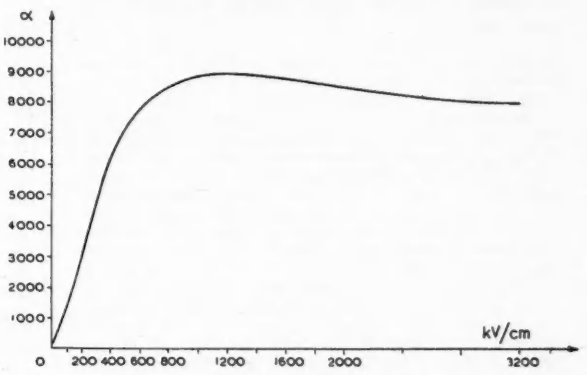

Fig. 2

$$n = n_o \frac{(\alpha-\beta)\, e^{(\alpha-\beta)d}}{\alpha-\beta\, e^{(\alpha-\beta)d}}$$

and the condition for breakdown :

$$\alpha = \beta \cdot e^{(\alpha-\beta) \cdot d}$$

where β is a function of the number of electrons liberated from the cathode by the positive ions and d the distance (gap) between the electrodes ; theoretically, n becomes infinite, and in practice is only limited by diffusion. As a result of the above, the time lag calculated by this method very considerably exceeds the experimental values.

There are various other theories applicable to the second part of phase I, the most important of which is due to Rogowski. This emphasizes the influence of positive space charges producing considerable deformations of the electric field between the electrodes. Hence the expression for the " electron avalanche " :

$$n = n_o\, e^{\int_o^x \alpha\,(\xi)\, d\xi}$$

Owing to the non-linear way in which α varies with the field strength, the presence of space charges will speed up the formation of the discharge. In addition, Rogowski took account of the ionisation of the gas by positive ions. Though this theory gave much improved results, it failed to explain very rapid discharges of the order of 10^{-8} sec, or short discharges under special conditions (high values for the product of pressure and width of gap).

A very recent theory put forward by Loeb and Meek, introduces the photo-electric effect of molecules excited by the impact of electrons. The molecules, about 5-10 times as numerous as the pairs of ions produced, emit, for 10^{-8} seconds, radiations of very short wave length which produce photo-electrons in the gas. The photo-electrons may be formed close to the initial avalanche or at a distance, when they will initiate secondary avalanches which may merge with the initial avalanche. Although this theory arrives at shorter times for the formation of discharges, it seems that it is still unsatisfactory for ultra-rapid discharges, especially for high values of pd.

A new theory, put forward by the author of this article, is described in the following. This theory takes for the first time account, not only of the positive charges left behind by the avalanche, but also of the positive charges created by, and carried in front of the latter. It will be seen that these positive charges assume very considerable values.

If, as a first approximation, it is assumed that the front of the avalanche is a plane whose negative surface density σ_e represents the whole negative charge of the avalanche, then the additional field strength on the axis of the latter, immediately in front of it, and acting in the same direction as a main field, will be $E_a = 2\pi\,\sigma_e$.

The negative charge of the avalanche is

$$Q_e = -\, n_o \epsilon\, e^{\alpha_k x_k},$$

where x_k represents the distance travelled from the cathode (Fig. 3), α_k a factor corresponding to x_k, ϵ the electron charge of $4{\cdot}8 \cdot 10^{-10}$ e.s.u., and n_o the number of initial electrons which, for simplification, will be assumed equal 1.

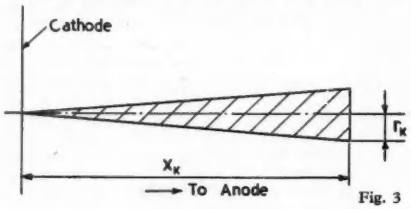

Fig. 3

On the other hand the positive charge left in the wake of the avalanche has a retarding effect on the latter; as a first approximation, only those positive charges will be taken into account, which are at a distance of not more than twice the diameter of the avalanche, and they will be reduced to the front of the avalanche. The resulting error is comparatively small. The positive charge thus reduced to the front of the avalanche is :

$$Q_1 = \epsilon \left(e^{\alpha_k x_k} - e^{0{\cdot}8\ \alpha_k x_k}\right)$$

and the total resulting charge therefore :

$$Q_r = \epsilon\left[-e^{\alpha_k x_k} + \left(e^{\alpha_k x_k} - e^{0{\cdot}8\ \alpha_k x_k}\right)\right] = -\epsilon e^{0{\cdot}8\,\alpha_k x_k}$$

whence :

$$E_a = -\frac{2\pi\,\epsilon\, e^{0{\cdot}8\ \alpha_k\, x_k}}{\pi\, r_k^2}$$

where r_k is the equivalent radius of the front of the avalanche at the point x_k, and always smaller than $0{\cdot}5\ x_k$; the additional field strength just in front of the avalanche is therefore at least :

$$E_a = -\frac{800\ \epsilon\, e^{0{\cdot}8\ \alpha_k\, x_k}}{x_k^2} \text{ e.s.u.}$$

The phenomena connected with the field strength are cumulative ; an increased field strength imparts a higher kinetic energy to the electrons and thereby increases their ionising power. This accelerates the formation of the avalanche, which in turn increases the field strength. The additional field created during the discharge may become not only equal, but of a higher order than the original field ; this explains the ultra-rapid discharges obtained in experiments.

In the following, a more detailed account is given of the mechanism, the basic principle of which has been explained in the foregoing. It is evident that if the motion of one group of electrons is greatly accelerated, the motion of another group will be retarded. The electron cloud forming the spearhead of the avalanche will undergo a distention and dislocation, a phenomenon which had hitherto been unexplained.

It is obvious that the electrons are not concentrated at the front of the avalanche, but more likely form an electron cloud. Since an assumption has to be made for the thickness of this cloud (this does not, however, influence the validity of the final theory) it will be assumed such as to allow the electrons contained therein, the number of free travels necessary to double the number of electrons by collision with neutral molecules. Thus the cloud will comprise twice as many electrons than positive ions. The negative charge of the electron cloud will therefore be :

$$Q_e = e^{\alpha_k\,(x_k - \delta_k)} = \frac{e^{\alpha_k\, x_k}}{2}$$

The cloud will be subject to three distinct forces, the original field E_o, the expansive forces due to internal charges, equivalent to a field E_e, and to a retarding field E_1 due to ionic charges in the tail of the avalanche (Fig. 4.).

The resultant force at the front face of the cloud is therefore :

$$E_{r\,(\text{front})} = E_o + E_{e\,(\text{front})} - E_{1\,(\text{front})}$$

and at the rear face :

$$E_{r\,(\text{rear})} = E_o - E_{e\,(\text{rear})} - E_{1\,(\text{rear})}$$

Assuming as a first approximation that the charges

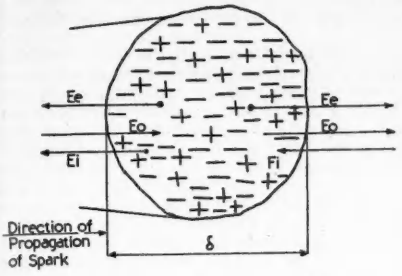

Fig. 4

are distributed over the avalanche in plane and parallel layers, while the number of positive ions found at the rear face of the avalanche for each distance $d\xi$ is known to be $\alpha\, e^{\alpha\xi}\, d\xi$, the following expressions can be derived for E_e, $E_{1\,(front)}$, and $E_{1\,(rear)}$:

$E_e = E_{e\,(front)} = E_{e\,(rear)} =$

$$= \frac{2\pi\epsilon}{\pi r_k^2} \int_o^{\delta_k} e^{\alpha_k(x_k - \delta_k)} \frac{1}{\delta_k} \left(1 - \frac{\delta_k - \xi}{\sqrt{r_k^2 + (\delta_k - \xi)^2}}\right) d\xi \quad (1)$$

$$E_{1\,(front)} = 2\pi\epsilon\alpha \int_o^{x_k - \delta_k} \frac{e^{\alpha\xi}}{\pi r^2} \left(1 - \frac{x_k - \xi}{\sqrt{r^2 + (x_k - \xi)^2}}\right) d\xi \quad (2)$$

$$E_{1\,(rear)} = 2\pi\epsilon\alpha \int_o^{x_k - \delta_k} \frac{e^{\alpha\xi}}{\pi r^2} \left(1 - \frac{x_k - \delta_k - \xi}{\sqrt{r^2 + (x_k - \delta_k - \xi)^2}}\right) d\xi \quad (3)$$

With $r_k = r_{max} = 0{\cdot}05\alpha_k$ equ. (1) yields :

$$E_{e\,(min)} = \frac{40\epsilon}{x_k\, \delta_k} e^{\alpha_k(x_k - \delta_k)} \frac{20\delta_k + \sqrt{400\delta_k^2 + x_k^2} - x_k}{20\,\delta_k + \sqrt{400\delta_k^2 + x_k^2}} \quad (4)$$

Equations (2) and (3) cannot be integrated directly and must be calculated for successive layers of the avalanche tail, α being assumed constant in each layer.

Due to different fields acting on the front and rear faces of the cloud, their respective velocities will differ, and the cloud will distend. At the same time the value of α increases, which causes a reduction in δ in inverse proportion. Thus, the rear face moves at times in the same direction as the front face, and at times in the opposite direction, because, as the electrons move back, the reverse field diminishes ; the original field gains the upper hand and the electrons move forward once more, producing the turbulence found in experiments.

Strictly speaking these results apply only to the electrons on the axis of the avalanche. Away from it the fields and speeds decrease, and the cloud must become ovoid, which is confirmed experimentally as regards the front face, which alone can be observed.

A rapid fall in acceleration is to be expected when the field strength reaches 1500 to 2000 kV/cm, when α begins to decrease.

Stabilization of the acceleration is reached at about $E_{r\,(front)} = 5000$ kV/cm corresponding to a velocity of the front face of the order of 4 to 5 $\times$ 10^8 cm/s. This limit is reached very rapidly, in about 3 to 4 $\times$ 10^{-9} seconds in a gap 1cm wide with 50 per cent over-voltage.

Although the theory developed above appears to be adequate to explain all experimental data available on ultra-rapid discharges, it is believed that also other effects play a part, in particular photo-electric phenomena not taken into account by this theory.

(c) *Phase* II.

The passage of the first avalanche, although making the initial discharge, is not by itself sufficient to establish the spark. For that, other avalanches must follow in unbroken succession, the electrons necessary to initiate the latter must be freed and, generally, the cross-section of the discharge must be increased to prepare the passage for the final current. These processes take place in the second and third phases ; they imply that positive ions have been active : either through ionisation at the surface of the cathode, or through ionisation by collision in or outside the space occupied by the avalanche. Assuming that both types of ionisation take place, which seems to be the general case, the following condition must be valid if the discharge is to continue :

$$k = \left[\beta + \frac{\int_o^d \gamma d\xi}{\int_o^d \alpha d\xi}\right] \cdot \left[e^{\int_o^d \alpha d\xi} - 1\right] \geqslant 1 \ldots (5)$$

This cannot be evaluated numerically as some authors have tried to do because of the uncertain values of β and γ, and because the photo-electric effects and the thermal ionisation have been neglected, both of which tend to promote the formation of the discharge. However, the above formula may be adequate to attempt a physical explanation of experimental data obtained on the formation of sparks.

If the discharge is initiated at a voltage only slightly exceeding the minimum striking voltage, the value of the ionisation factor k will, only slightly, exceed unity. The discharge will be continuous, the current intensity increasing, and the voltage across the gap directly decreasing (due to external series impedances) to a value corresponding approximately to the arc voltage ; the phases II and III will therefore merge.

It may, however, happen that k drops below unity during the process, especially if the voltage applied is near the minimum striking voltage ; or if the series impedances are large, or if the discharge takes place in a rarefied gas or if the capacity of the source is small. In this case the current ceases to increase and the voltage settles at 300 volt (the "normal" cathode-fall) or at a considerably higher value (the "abnormal" cathode-fall). This however, never happens if the discharge takes place in a compressed gas

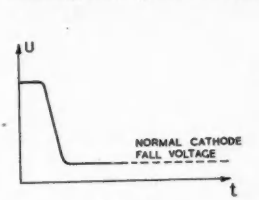

Fig. 5a

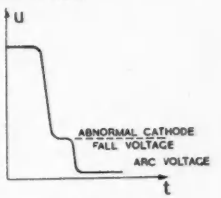

Fig. 5b

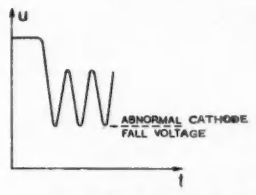

Fig. 5c

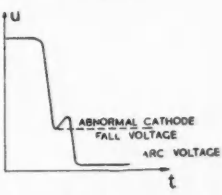

Fig. 5d

If the process halts at the "normal" cathode-fall voltage and the circuit is such as to limit the resulting current to a fraction of one ampere (at atmospheric

pressure), conditions may become stable and phase III may not develop (Fig. 5a).

On the other hand, if the voltage halts at the " abnormal " cathode-fall value, a highly unstable condition results, due to the considerable power requirements obtaining in this case. If the capacity of the source is adequate to meet them, the extraction of ions from the cathode by the electric field and thermal ionisation will make the ionisation factor k greater than unity and the voltage will drop sharply to the arc voltage (Fig. 5b). If the capacity of the source is inadequate, the discharge will tend to cease and will do so, either completely (interrupted discharge) only to begin again as soon as the returning voltage reaches a value in the neighbourhood of the initial striking voltage (Fig. 5c), or partially, if the voltage returns slowly while k becomes again greater than unity. In the latter case the discharge will generally come to an end by the voltage dropping sharply to the arc voltage (Fig. 5d). The four distinct types of discharge, as described above, have actually been observed by several authors.

The duration of phase II, assuming only ionisation by positive ions on the surface of the cathode, will be of the order of 10^{-6} sec ; if also ionisation by collision and photo-electric effects is taken into account, the theory gives a duration of the order of 10^{-9} seconds, which is in agreement with actual measurements.

(d) *Phase* III.

As has been shown, the voltage of the gap may, under certain conditions, fall below the " normal " cathode fall voltage to what is called the " arc voltage " (Figs. 5b, d). The conditions referred to in the foregoing, however, though necessary, are not sufficient to produce this effect and additional causes may be found either in the emission of electrons by the cathode (due to a concentration of field strength near the surface, to photoelectric effects, or to thermionic emission), or in the formation of ions and electrons in the gap (due to thermal ionisation or photo-electric effect).

As the field, when concentrated in the cathode space, may reach intensities of the order of 1000 kV/cm, extraction of electrons from the cathode plays a very important part, especially in initiating phase III, the concentration of the field being higher at the onset of this phase. On the other hand, the relative importance of thermionic emission is greater at the end of the phase, the more so, the longer the duration of this phase. It has been calculated that thermionic emission in the gas forming the cathode space begins in less than 10^{-9} secs. The temperature rise of a copper cathode in the same time is found to be only about 50 deg. C, which is practically negligible and which demonstrates the interesting fact that in the case of an ultra-rapid striking of a spark, the cathode remains cold. Likewise it is unlikely that the bombardment of the cathode by positive ions can play any important role, since, in the short time of about 10^{-8} seconds scarcely any ions will reach the cathode.

On the other hand, the photo-electric effect seems certain to shorten the duration of phase III, but a quantitive assessment of its influence is difficult.

From the above deductions, it will be seen that the duration of phase III can be as short as a fraction of 10^{-9} seconds, and that the total duration of the forming of the spark can be of the order of 10^{-9} seconds, which has been confirmed by experiment.

III. THE ESTABLISHED STATE.

(a) *General.*

It may seem a contradiction in terms to refer to an " established state " of a spark, a spark being by definition a transcient phenomenon leading to an arc ; however, it may be permissible to do so if dealing with phenomena of short duration, as thermionic emission from the electrodes is practically non-existent, and as there is no rigid limit between what may be considered the end of the spark and the beginning of the arc.

The established states of sparks differ considerably depending on whether the spark is caused by d.c or a.c. and, in the latter case, on the frequency. In the case of d.c. or of a.c. of low or medium frequencies (up to about 10^4 c/s.), the characteristics of the spark seem to be determined almost exclusively by the thermal balance of the energy supplied to, and dissipated in the electrodes and in the surrounding space. In the case of d.c., there is a constant equilibrium in the established state between the supplied and the dissipated energy. The temperature of the spark remains constant and so does the resistance and the voltage across the gap. In the case of a.c. of low and medium frequencies, however, there is no permanent equilibrium, as, due to the periodical passage of the current (which determines the energy supplied) through zero on one hand, and the thermal hysteresis (which influences the dissipation of energy) on the other, the energy balance will fluctuate between positive and negative. Consequently also the temperature, the resistance of the circuit, and the potential across the gap all fluctuate in the same rhythm ; there is a voltage rise before and after each passage of the current through zero, the latter being the more pronounced because of thermal hysteresis (Fig. 6). On the

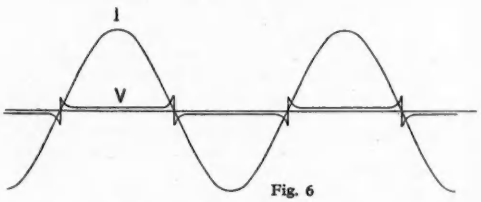

Fig. 6

other hand, due to the same thermal hysteresis, the temperature fluctuations tend to decrease if the frequency increases, so that, as far as the electric resistance is concerned, a spark caused by a very high frequency (10^6 c/s or more) could be expected to behave exactly like a spark caused by d.c. This, however, is not the case, because the inertia of the ions and electrons becomes an important factor at very high frequencies, at which also the equivalent resistance of the spark increases considerably.

(b) *High and Very High Frequencies.*

In the following, the results of experimental investigations into the behaviour of sparks at high and very high frequencies are given and a theory is put forward which may provide a physical explanation. At very high frequencies the equivalent voltage of the spark, i.e. the voltage which, when multiplied with the r.m.s. value of the current gives the energy dissipated in the spark, increases rapidly with the frequency. It is, for instance (for millimetre sparks in air at atmospheric pressure) of the order of 50 volts at 50 c/s, but exceeds 1000 volts at 10^8 c/s.

This increase is due first to the voltage peak necessary for the re-formation of the spark which seems to increase steadily with the frequency ; and second to the lengthening of the formative period, which seems to

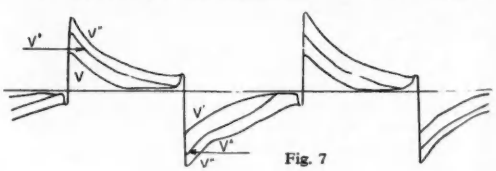

Fig. 7

follow a similar law. Consequently the duration of the voltage of low amplitude across the gap decreases relative to the duration the high voltage (Figs. 6 and 7) until, from a certain frequency onward, the trough of low voltage disappears altogether, the re-formative period exceeding the duration of the half-cycle (Fig. 7).

Since it is essential, for maintaining a sensibly constant voltage across the gap, that the cross-section of the spark should change, the first step in the theoretical explanation must be to examine these changes. This is done on the basis of the "electron equilibrium" as previously explained (see equation 5) which will be more positive on the axis than on the edge of the spark where electrons are lost through diffusion. The distribution of k is indicated in Fig. 8.

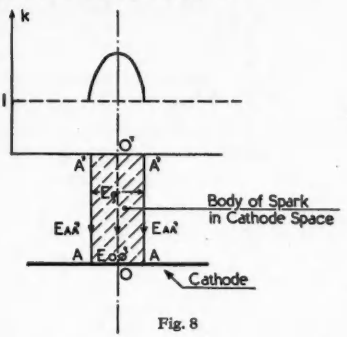

Fig. 8

To each value of k there exists a corresponding axial field strength, and the distribution of the field will be so that it will be stronger on the axis (E_{oo}) than on the periphery (E_{aa}). This in turn necessitates a radial field E_ρ which seems to play a very important part in the adjustment of the cross-section of the spark to the intensity of the current, as well as in the interruption of the spark.

Thus, if the current increases, while the cross section of the spark, and therefore the resistance remains constant, E_{oo}, E_{aa} as well as E_ρ increase and consequently the cross-section tends to increase until a new equilibrium condition is established ; this means smaller axial field intensities, for, with the ratio of area to perimeter increasing, also the ratio of number of electrons produced to electrons lost by diffusion increases, which accounts for the negative characteristic of the spark (decrease of voltage for increase of current).

If the current decreases, the process is reversed, the axial and radial field intensities increase until, at a critical area, the spark is rendered unstable by E_ρ and finally extinguished, before the current actually reaches zero.

However, the cross section of the spark varies in step with the current only as long as the rate of variations of the radius corresponding to this cross-section is small compared with the velocity of the electrons in the radial direction, i.e. as long as :

$$\frac{d\rho}{dt} \ll v_{e\rho}$$

This condition can be transformed into :

$$\frac{fS}{\rho} \ll v_{e\rho}$$

where f is the frequency and S the maximum cross-section of the spark.

This can be further developed into :

$$f \ll \frac{2 \cdot 10^7 \sqrt{\rho}}{S}$$

and

$$f_{max} = \frac{2 \cdot 10^6 \sqrt{\rho}}{S}$$

From this it follows that the cross section of the spark ceases to vary in sychronism with the current when the frequency reaches values in the region of 10^6 c/s. This limit is the lower, the lower the ratio ρ/S.

The radial inertia of the spark evidently causes a loss of symmetry in the voltage curve of one half-cycle, with respect to the ordinates $\pi/2$ or $3\pi/2$. This becomes the more pronounced, the higher the frequency, and is further increased by the growing divergence between the values of the peaks of extinction and re-formation.

On considering the voltage at the moment of extinction, it appears that the inability of the space between the electrodes to recover its dielectric strength sufficiently quickly must limit it to less than the minimum breakdown potential.

On the other hand conditions are quite different for the point of re-formation. When the spark is extinguished a barrier of positive ions is left in front of the cathode, and of electrons in front of the anode. In the new half-cycle this anode becomes the cathode while the old cathode becomes the anode. If a current is to flow in the reverse direction in the new half-cycle, the intensity of the field will be adequate to : (i) nullify the established field and the charge of the positive ions (component V_1), (ii) carry the existing electrons to the new anode V_2 and (iii) extract new electrons from the new cathode in 10^{-10} seconds or less, V_3. The first component V_1 has obviously the effect of shifting the phase between current and voltage. On the other hand V_2 and V_3 determine the re-forming voltage. V_3 must be in the region of 1000 volts for frequencies of the order of 10^8 c/s and grows slowly with the frequency. On the other hand V_2 increases rapidly with the frequency, since it depends on the time available to transport the electrons to the anode, as well as on the rate of the voltage increase.

It can be shown that :

$$V_2 = \left(\frac{2\omega_0 U}{k}\right)^{\frac{2}{3}} d_2$$

where $\omega_0 = 2\pi$ times frequency of the open circuit, U = amplitude of the applied e.m.f., d_2 = distance between electrons and anode. k is obtained from the equation: velocity $= k \sqrt{}$ field strength.

It can be seen from the above equation that the amplitude at re-formation increases very rapidly with the frequency producing the lack of symmetry which had been found experimentally (Fig. 7). For instance, with $U = 30000$ volts and $d \approx 10^{-3}$ cm, V_2 is negligible for $f = 10^6$ c/s but about 2000 volts for $f = 10^9$ and about 9000 volts for $f = 10^{10}$ c/s. Under the same conditions the equivalent voltage is about 650 volts for $f = 5.10^7$ c/s, rising to about 2000 volts for $f = 10^9$ c/s.

This marked voltage-drop gives high attenuation factors for the circuit containing the spark, which explains why it has been practically impossible to produce oscillations which were not heavily damped, in gases at atmospheric pressure.

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THE DIMENSIONING OF AUTOMOBILE RADIATORS.

By B. LATTANZI. (From *L'Ingegnere*, Vol. 20, No. 6, June, 1946, pp. 429-432, 7 illustrations.)

THE development in recent years of liquid-cooled aircraft engines has given a powerful impetus to the perfection of engine radiators. There is no doubt that radiators of this type will be applied to post-war automotive designs where they will improve the cooling effect and reduce frontal area, weight, and cost of the cooling surface. Radiators can be classified under three headings, namely :

(a) Air tube coolers. (b) Cellular coolers. (c) Water tube coolers.

War time requirements also led to the introduction of special types such as the vertical plate type with horizontal fins and others. But it is in the nature of things that the thermodynamic advantages obtainable with such designs cannot be utilized in commercial practice owing to high production costs.

The wartime shortage of copper experienced in Italy and Germany forced the radiator designers of these countries to resort to the use of galvanized iron sheet for the extended surfaces, or to construct the entire radiator of aluminium. In this connection it will be of interest to consider the overall heat transfer coefficient obtaining between the cooling air and the water. This coefficient is given by :

$$\alpha = \frac{1}{\frac{1}{\lambda_1} + \frac{s}{\lambda} + \frac{1}{\lambda_2}}$$

Here the term s/λ is negligible for small wall thicknesses, no matter which metal is used, excepting radiators with extended surface the presence of which is equivalent to a considerable increase in wall thickness. In Germany various types of air tube radiators and water tube radiators completely built of aluminium were investigated. In the water tube type the extended surfaces were affixed to the water tubes by means of synthetic resins mixed with aluminium powder.

Leaving aside wartime emergency designs, competition of many years standing has ended with the acknowledgment of the superiority of the water tube type provided with extended surfaces, this superiority having been proven by exhaustive aerodynamic tests. With tubular radiators it is possible to achieve high values for the ratio of free air flow area to cross sectional area of the apparatus—and this for small hydraulic diameters of the air passages. Owing to this fact high cooling effects are achieved with a shallow apparatus of low weight and small power absorption. This type also possesses the advantage that it leads to a reduction in incrustations. The length of soldered seams is also reduced which is a particularly weak point of the honeycomb type radiator which has some 9 metres of soldered seams per dm² of frontal area.

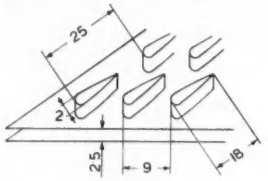


Fig. 1

In testing various types, W. Linke* found that the design shown in Fig. 1 gives the best results. This type which embodies streamlined tubes with successive rows arranged in alignment, is built in elements of 125 mm width, the weight without water being 1·35 kg per dm² and the weight of its water contents being 0·16 kg per dm². The ratio of free air flow area to total cross section is 0·72. The characteristics of this design, given in Fig. 2, as function of the air velocity v are based upon the equation :

$$Q = vs\,\Delta t\, c_p\, \gamma = 3600\, c_p\, \gamma\, \Theta\, s\, r\, v$$

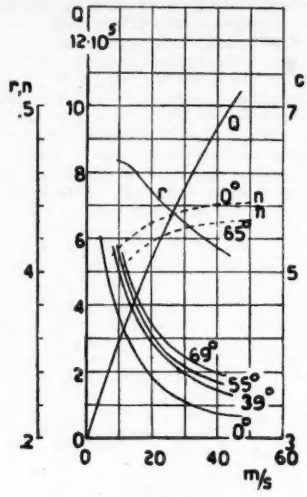


Fig. 2

where Q is the amount of heat dispersed, it being $r = \Delta t/\Theta$, where the factor r represents the degree of air temperature rise. This factor expresses the ratio of the heat actually removed to the amount of heat which would be removed if the air temperature at the radiator exit would equal the temperature of the liquid. The factor :

$$n = \frac{1}{\sqrt{1 + c_x}}$$

signifies the permeability of the radiator, where c_x is the coefficient of aerodynamic resistance based upon the loss in air pressure across the radiator. Note should be taken that the coefficient c_x increases with the temperature according to the empirical relationship

$$c_x' \cong c_x \left(1 + 1{\cdot}4\,\frac{\Delta T}{T_1}\right)$$

where T_1 is the absolute temperature of the heated air at the outlet from the radiator. This rise in the value for c_x is due to the increase in pressure drop caused by the increased velocity of the heated air. In order to convey an idea of the variation of the aerothermodynamic characteristics of the radiator with the depth of the apparatus, Fig. 3 is included which shows the test results obtained with radiators of 125, 175, and 225 mm depth respectively.

It will be of interest to investigate the variation in the heat transmitted to the air for different construction materials such as copper, aluminium, or iron. If α is the overall heat transfer coefficient of the entire apparatus, Q_d is the heat dispersed by the direct surface S_d, and Q_l is the heat dispersed by the indirect (extended) surface S_l, we have :

$$Q = Q_d + Q_l = S_d\, \alpha\, \Theta_m + \eta_2\, S_l\, \alpha\, \Theta_m$$

where Θ_m is the temperature difference between the

*Jahrbuch d. Deutsch. Luftfahrtforschung, 1940.

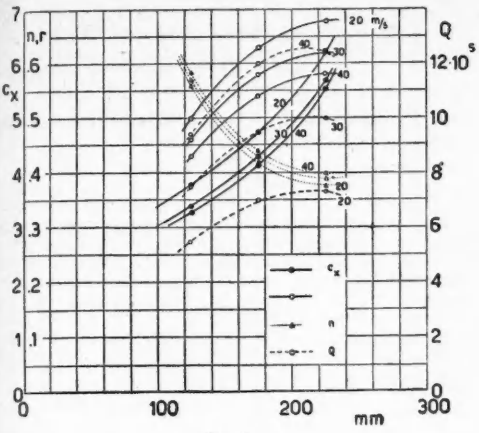


Fig. 3

temperature of the direct-heated surface and the mean temperature of the air, and η_2 is the efficacy of the extended surface. The latter is a function of the length b of the fins and of their thickness δ, of the thermal conductivity λ of the material, and of the coefficient of heat transfer. This functionality is given by :

$$\eta_2 = \sqrt{\frac{2\lambda\delta}{b^2 a}} \operatorname{tg} \frac{1}{\sqrt{\frac{2\lambda\delta}{b^2 a}}}$$

For $\delta = 0{\cdot}11$ mm and $b = 11$ mm of fins of copper, aluminium, or iron, the respective values are $\eta_2 = 0{\cdot}92$, $0{\cdot}81$, and $0{\cdot}59$, assuming a heat transfer coefficient of 200 cal/h, m², deg. C. making $S_d = S.t$ and $S_t = S.u$, we have

$$Q = S\,\alpha\,\Theta_m\,(t + u\,\eta_2)$$

This allows to determine the factor r as

$$r = 1 - e^{-1/2\,\eta_1\,(t + u\,\eta_2)\,c_x\,f^2}$$

where η_1 is a correction factor given as $0{\cdot}8 \leqslant \eta_1 \leqslant 1{\cdot}10$.

The dimensions of automobile radiators have hitherto been chosen on the basis of such empirical data as are available. In up to date automobile technique, however, the radiator must be considered as a heat dispersing body situated in a diverging-converging duct, that is to say, as a streamlined radiator of the type shown in Fig. 4. The amount of heat which will be abstracted from the water per hour is given by

$$Q = 3600\,\gamma\,c_p\,\Theta\,s\,r\,v_1$$

where v_1 is the air speed in front of the apparatus at the point where the air enters the cooling surface. By introducing the coefficient of discharge $m = v_1/v$, it can also be written

$$Q = 3600\,c_p\,\gamma\,\Theta s\,v\,m\,r$$

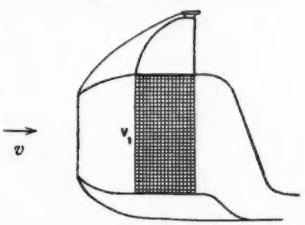


Fig. 4

and by applying Bernouilli's theorem we find

$$m = \frac{1}{\sqrt{1 + c_x' + \frac{T_2}{T_1}\left(f^2\frac{s^2}{s_3^2} - 1\right)}}$$

where c_x' is the aerodynamic resistance of the radiator element, T_2 and T_1 are the absolute temperatures at the radiator inlet and outlet respectively, s_3 is the sectional area of the air inlet, and f is the ratio of the free flow area to the total cross section of the radiator. The coefficient $\eta^* = \eta_1\,(t + u\,\eta_2)$ has been found to vary with the air velocity. For air velocities in excess of 20 m per second it is given by the expression $\eta^* = 0{\cdot}8 - 0{\cdot}0053\,v_1$ which applies to copper radiators of the water tube type. The depth of the radiator is only of slight influence in this connection. The heat dispersal Q will attain a maximum if the product mr also attains a maximum.

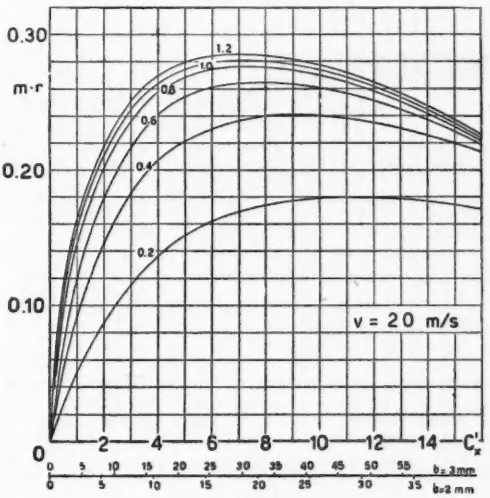


Fig. 5

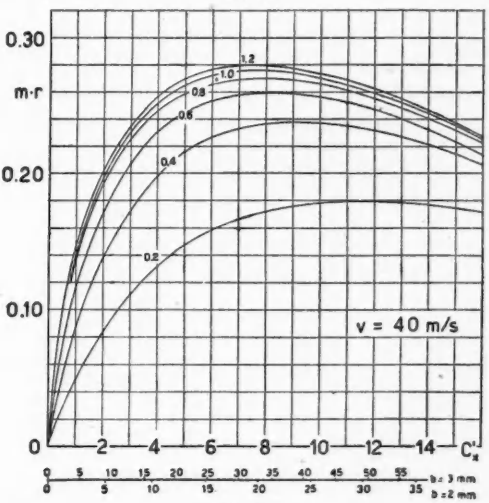


Fig. 6

Therefore

$$\frac{dmr}{dc_x'} = \frac{d}{dc_x'} \frac{1}{\sqrt{1 + c_x\left(1 + 1{\cdot}4\frac{\Delta T}{T_1}\right) + \frac{T_2}{T_1}\left(f^2\frac{s^2}{s_2^2} - 1\right)}} \times \left(1 - e^{-\frac{1}{4}(0{\cdot}8 - 0{\cdot}0053 v_1) f^2 c_x (1 + 1{\cdot}4\Delta T/T_1)}\right) = 0$$

The numerical values for mr computed for a given case with the use of the expression given above are charted in Figs. 5 and 6 for air velocities of 20 m per second and 40 m per second respectively. The following numerical values for the various factors were used : Temperature of the entering air 30 deg. C = 303 deg. C. abs., $\Delta t = 5$ and $\Delta t = 10$ deg. C., $f = 0{\cdot}75$. The results for the two Δt values are practically identical. The influence of the air velocity was found to be small. The optimum frontal area s of the radiator can be found by introducing into the equation for Q the appropriate value for $m \cdot r$ as given in Figs. 5 and 6. The depth l of the radiator can be determined on the basis of Blasius' law for turbulent friction

$$c_f = 0{\cdot}316\, R^{-0{\cdot}25} = c_x' \frac{b}{l} f^2$$

where R represents the Reynolds number of the radiator under consideration, so that

$$R = \frac{vb}{\nu}$$

where b is the distance between the fins, and ν is the kinematic viscosity which, for air temperatures from 35 to 40 deg. C., approximates to 0·15 cm²/sec. It will be noted that Figs. 5 and 6 are based upon two alternative values for b, namely $b = 3$ mm and $b = 2$ mm. It will readily be understood that in the case of automobiles, in which the factor s/s_3 approximates to unity, favourable values for mr will obtain with the radiator depth ranging between 7 and 10 cm.

SWITZERLAND

ALTERNATING CURRENT PREMAGNETIZATION OF FERROMAGNETIC CORES.

By R. RISCH. (From *Brown Boveri Mitteilungen*, Vol. 33, No. 6/7, June/July, 1946, pp. 129-133, 7 illustrations.)

INTRODUCTION.

PRE-MAGNETIZATION by means of alternating current increases the effective permeability of a transformer, thus allowing (for a given magnetizing current) a smaller cross-section or fewer turns. Alternatively, for the same cross-section and number of turns, a smaller magnetizing current can be used, resulting in smaller current losses and better power factor.

In the usual arrangement (Figs. 1a and 1b) the core K is in two parts. The two magnetically equal sections are pre-magnetized, equally but in opposition directions by the special winding 3, so that no e.m.f. is induced in the other windings. 1 is the primary, consisting in Fig. 1a of two parts with the same number of turns, each wound on one of the core sections. In Fig. 1b, winding 1 is shown as a single winding embracing both cores ; 2 is the secondary and is shown in Fig. 1a as made up of two parts each on a core section, while in Fig. 1b it embraces both cores.

In considering the superposition of the premagnetization and the normal magnetization it is convenient to consider three cases :—

1. Premagnetization in phase with normal magnetization.
2. Premagnetization 90 deg. out of phase with normal magnetization.
3. Arbitrary phase difference.

The premagnetization is assumed large relative to the normal magnetization, for only in this case can the permeability be greatly increased.

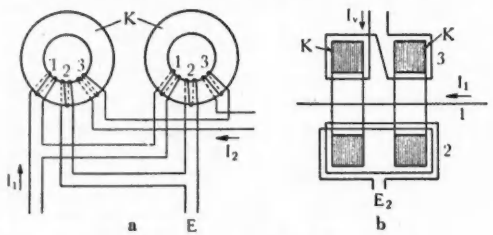


Fig. 1. Schematic diagram.

K = Core.
1 = Winding for normal magnetization.
2 = Secondary winding.
3 = Premagnetization winding.

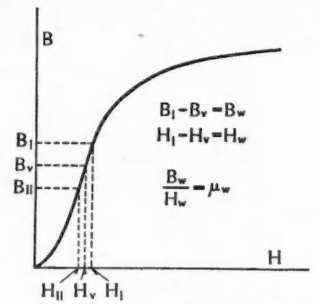


Fig. 2. Magnetic relationships (Premagnetization in phase with normal magnetization.)

H_v = Field-strength (Premagnetization).
B_v = Flux Density (Premagnetization).
H_w = Normal field.
B_w = Normal flux density.
H_I and B_I = Field-strength and flux density in core I.
H_{II} and B_{II} = Field-strength and flux density in core II.
u_d = Differential permeability.

PREMAGNETIZATION IN PHASE WITH NORMAL MAGNETIZATION.

If $\mathfrak{J}_1$, in winding 1 is in phase with the premagnetizing current $\mathfrak{J}_v$ in winding 3, the ampere-turns of $\mathfrak{J}_1$, of the one core-section are added to those of $\mathfrak{J}_v$, while they are subtracted from the other core-section, due to the windings being reversed. If the field-strength due to $\mathfrak{J}_v$ is H_v, while that due to $\mathfrak{J}_1$ is H_w, the resultant field-strength in core I is $H_I = H_v + H_w$, and in core II, $H_{II} = H_v - H_w$. Since H_v and H_w are in phase, H_I and H_{II} are each in phase with H_v. H_I and H_{II} give, according to the B/H curve shown in Fig. 2, peak values of flux density B_I and B_{II}. B_I and B_{II} produce the two e.m.f.'s E_I and E_{II}, whose difference is the secondary e.m.f. of the whole apparatus. B_I and B_{II} are not generally in phase since conditions are different when $H = H_I$, and when $H = H_{II}$. Yet if H_w is small relative to H_v, then H_v, H_I and H_{II} differ little, and B_v, B_I and B_{II} can be taken as in phase. This

implies that the secondary e.m.f. E_2 and the premagnetization e.m.f. E_v are also in phase, hence $B_I - B_v = B_v - B_{II}$ and $B_I - B_{II} = 2(B_I - B_v) = 2B_w$, if B_w is the effective induction, due to the effective field-strength. The secondary e.m.f. is then

$$E_2 = E_I - E_{II} = 4{\cdot}44 \,.\, f \,.\, z_2 \,.\, F' (B_I - B_{II}) \,.\, 10^{-8} \;\ldots\; (1)$$

where z_2 = number of secondary turns on each core
F' = cross-section of a core

i.e. $E_2 = 4{\cdot}44 \,.\, f \,.\, z_2 \,.\, F \,.\, B_w \,.\, 10^{-8} \;\ldots\; \ldots \quad (2)$

where F = cross-section of both cores together.

Equations (1) and (2) are true whatever the phase relationship, but in (1) $(B_I - B_{II})$ must be the vector difference.

From Fig. 2.

$$\frac{B_w}{H_w} = \mu_d$$

where μ_d represents the differential permeability dB/dH. Hence, when the two currents are in phase, the effective permeability equals the differential permeability at the operating point on the B/H curve.

PREMAGNETIZATION PERPENDICULAR TO NORMAL MAGNETIZATION.

If H_w is 90 deg. out of phase with H_v and small relative to it, the addition of H_w to H_v does not alter the amplitude of magnetization of the core. The same is true if H_w is subtracted from H_v. In magnitude then $H_I = H_{II} = H_v$ and $B_I = B_{II} = B_v$. The secondary e.m.f. appears only because B_I and B_{II}, and E_1 and E_{II} are out of phase.

Since H_w is small relative to H_v, here too the angles between B_v and H_v, B_I and H_I, B_{II} and H_{II} are nearly equal and the triangles in the B– diagram are similar to those in the H– diagram. Hence the effective permeability

$$\frac{B_w}{H_w} = \frac{B_v}{H_v} = \mu_t.$$

where μ_t represents the total permeability B/H. Because of the similarity of triangles also the currents and e.m.f.'s are 90 deg. out of phase.

ARBITRARY PHASE DIFFERENCE.

In this case, the angle ψ_b between B_v and B_w is not equal to the angle ψ_h between H_v and H_w, hence two different expressions represent the effective permeability μ_w in terms of ψ_b and of ψ_h. Fig. 3 represents conditions for one core-section only. Since H_w and B_w are small relative to H_v and B_v the angle β between B_I and H_I is taken equal to that between B_v and H_v.

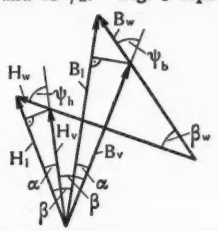


Fig. 3. **Vector diagram (Arbitrary phase difference).**
ψ_h or ψ_b = Phase-angle of premagnetization.
β = Angle of loss at operating point.
β_w = Apparent angle of loss.

Since α is small

$$B_w = \sqrt{(B_I - B_v)^2 + B_v^2 \alpha^2}.$$

$$\alpha = \frac{H_w \sin \psi_h}{H_v} \text{ and } H_I - H_v = H_w \cos \psi_h.$$

Hence $B_I - B_v = \mu_d (H_I - H_v) = \mu_d H_w \cos \psi_h$

$$B_v \,.\, \alpha = \mu_t H_v \,.\, \frac{H_w \sin \psi_h}{H_v} = \mu_t H_w \sin \psi_h.$$

$$B_w = \sqrt{\mu_d^2 H_w^2 \cos^2 \psi_h + \mu_t^2 H_w^2 \sin^2 \psi_h}$$

$$\frac{B_w}{H_w} = \mu_w = \sqrt{\mu_d^2 \cos^2 \psi_h + \mu_t^2 \sin^2 \psi_h} \qquad (3)$$

giving effective permeability μ_w in terms of ψ_h.

Similarly $H_w = \sqrt{(H_I - H_v)^2 + H_v^2 \alpha^2}$

$$H_I - H_v = \frac{B_I - B_v}{\mu_d} = \frac{B_w \cos \psi_b}{\mu_d}$$

$$H_v \alpha = \frac{B_v}{\mu_t} \,.\, \frac{B_w \sin \psi_b}{B_v} = \frac{B_w \sin \psi_b}{\mu_t}$$

$$H_w = \sqrt{B_w^2 \frac{\cos^2 \psi_b}{\mu_d^2} + B_w^2 \frac{\sin^2 \psi_b}{\mu_t^2}}$$

$$\frac{B_w}{H_w} = \mu_w = \sqrt{\frac{1}{\dfrac{\cos^2 \psi_b}{\mu_d^2} + \dfrac{\sin^2 \psi_b}{\mu_t^2}}} \;\ldots\; (4)$$

giving effective permeability μ in terms of ψ_b. For the limiting cases ψ_b, $\psi_h = 0$ deg. and ψ_b, $\psi_h = 90$ deg. equations (3) and (4) reduce to the values previously obtained, $\mu_w = \mu_d$ and $\mu_w = \mu_t$ respectively.

THE ANGLE BETWEEN B_w AND H_w

μ_w represents a relation between B_w and H_w, and the other relation, the angle between B_w and H_w, β_w is of interest, representing the drain on the source supplying winding 1 when winding 2 is on open circuit.

From Fig. 3.

$$\beta + \psi_h = \beta_w + \psi_b$$

or $$\beta_w = \beta + \psi_h - \psi_b \;\ldots\; \ldots \quad (5)$$

Eliminating μ_w from (3) and (4) gives the relation between ψ_h, ψ_b, μ_d and μ_t

$$\sin \psi_h = \frac{\sin \psi_b}{\sqrt{\left(\dfrac{\mu_t}{\mu_d}\right)^2 + \left[1 - \left(\dfrac{\mu_t}{\mu_d}\right)^2\right] \sin^2 \psi_b}} \qquad (6)$$

or $$\sin \psi_b = \frac{\sin \psi_h}{\sqrt{\left(\dfrac{\mu_d}{\mu_t}\right)^2 + \left[1 - \left(\dfrac{\mu_d}{\mu_t}\right)^2\right] \sin^2 \psi_h}} \qquad (7)$$

Thus if the magnitude and phase of the premagnetization are given, then μ_t and μ_d and the angle β can be determined from the quality of the iron, and either ψ_h or ψ_b can be calculated according to whether the phase of the current or of the e.m.f. is given. Equations (3) and (4) then give μ_w, equations (6) or (7) give ψ_h or ψ_b and (5) gives the angle β_w in terms of β, ψ_h and ψ_b. Thus if B_w is given, the phase and magnitude of H_w is derived, and *vice versa*.

VECTOR DIAGRAM OF THE PREMAGNETIZED TRANSFORMER.

Fig. 4 is based on these considerations. The load and the resistive and inductive impedance of the secondary winding, with the secondary current $\mathcal{J}_2$ give the secondary terminal voltage U_2, the resistive and inductive voltage e_r and e_s drops, and hence the induced e.m.f. E_2. At right angles to this is B_w, obtainable from equation (2). B_v makes an angle ψ_b to B_w. The angle β and the total permeability at the operating point give H_v from

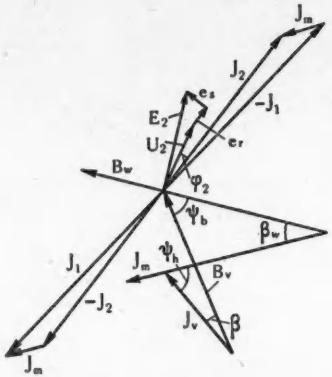


Fig. 4. Vector diagram of the premagnetization transformer.

$\mathfrak{J}_2$ = Secondary current.
φ_2 = Phase difference of load.
U_2 = Secondary terminal voltage.
e_r = Ohmic drop in secondary.
e_s = Inductive voltage drop in secondary.
E_2 = E.M.F. in secondary.
B_w = Normal flux density.
B_v = Premagnetization flux density.
β = Angle of loss at operating point.
$\mathfrak{J}_v$ = Premagnetizing current.
β_w = Apparent angle of loss.
$\mathfrak{J}_m$ = Magnetizing current.
ψ_b or ψ_h = Phase angle of premagnetization.
$\mathfrak{J}_1$ = Primary current.

B_v and also give $\mathfrak{J}_v$. Equ. (4) and (6) yield μ_w and ψ_h, and H_w, or $\mathfrak{J}_m$, the magnetizing current in winding 1 corresponding to H_w. $\mathfrak{J}_1$ and the angle of loss δ are obtained after reducing $\mathfrak{J}_2$ and $\mathfrak{J}_m$ to the same number of turns.

EXPERIMENTAL RESULTS.

Measurements confirm the theory. Voltages were generally made sinusoidal, distortion due to non-linearity of the B/H Curve appearing in the fields and magnetizing currents. The variation of the total and differential permeabilities, μ_t and μ_d with flux density were obtained from the B/H curve. The effective permeability was measured for several angles ψ_b and corresponding values calculated by equ. (4) for the same angles. Agreement was very satisfactory (Fig. 5).

In Fig. 6 variation of β_w with ψ_b for various premagnetizations B_w is shown in full line, corresponding calculated results being shown dotted for one value of B_v. Agreement is not so good for β_w as for μ_w, which is probably due partly to the assumption of equal angles between H_I and B_I, H_{II} and B_{II} and between H_v and B_v, and partly to the assumption that all quantities in the vector diagram are sinusoidal, while in fact the field-strengths are distorted.

It is noteworthy that β_v can be negative. This implies that B_w leads H_w and a flow of energy from the premagnetizing source to that supplying winding 1 occurs. Theory agrees with this only for large B_v and positive ψ_b. For large B_v, μ_t is greater than μ_d. ψ_h is less than ψ_b equ. (6), and for large B_v, β is known to be small. Under these conditions β_w can be negative according to equ. (5).

In the theory there is no dependence of μ_w on B_w unlike the case when premagnetization is absent. This has been verified experimentally, when it appeared that at least until $B_w = B_v/2$, μ_w was almost constant. This straight line inductance of premagnetized iron cores is well known to radio engineers.

SUMMARY.

Equations are derived by which the effective permeability μ_w and the phase angle β_w between B and H a.c. premagnetization can be calculated, given the magnetic qualities of the core and the phase of premagnetization. It is shown that when premagnetization and normal magnetization are in phase μ_w is the differential permeability dB/dH ; but if they are 90 deg. out of phase, μ_w is the total permeability B/H. In other cases, values lie between these extremes. Experiment shows exact agreement with theory for μ_w. For β_w there is qualitative agreement, but some quantitative differences exist, probably due to the assumption that all quantities varied sinusoidally, which is not exactly true.

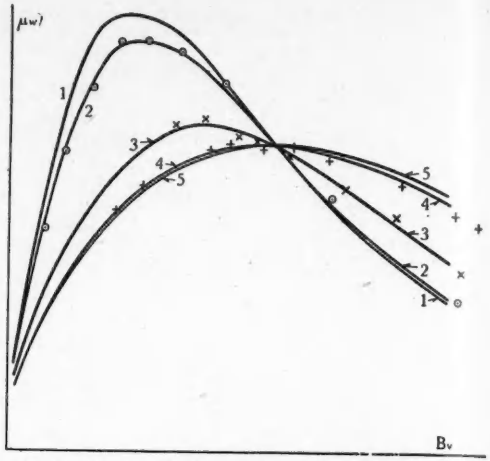


Fig. 5. Curves relating effective permeability and premagnetization flux density for various phase differences between normal and premagnetization.

1 = Theoretical curve for 0 deg.
2 = Calculated " " 15 deg.
3 = " " " 50 deg.
4 = " " " 80 deg.
5 = Theoretical " " 90 deg.
• = Experimental values for 15 deg.
× = " " " 50 deg.
\+ = " " " 80 deg.

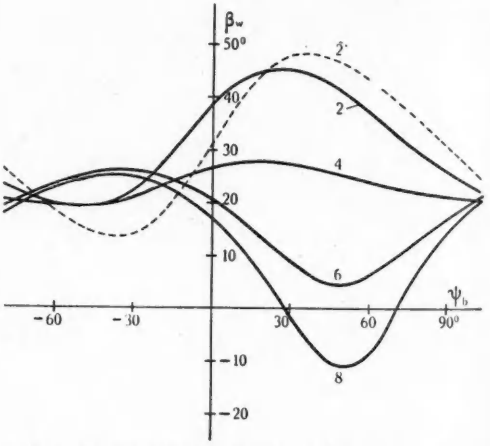


Fig. 6. Curves of β_w plotted against ψ_b for various values of premagnetization flux density.

β_w = Phase angle between normal field and normal flux.
ψ_b = Phase angle between premagnetizing flux and normal flux.
2' = Calculated curve for 2000 gauss.
2 = Measured " " 2000 "
4 = " " " 4000 "
6 = " " " 6000 "
8 = " " " 8000 "

Positive ψ_b means normal flux leads premagnetizing flux.

STATIC BALANCING OF TURBINE WHEELS.

By P.-A. ROUBINET. (From *Mécanique*, Vol. 30, No. 333, May 1946, pp. 121-124, 3 illustrations.)

THE oscillation method of statically balancing turbine wheels is well known to turbine manufacturers but rarely adopted by turbine users who, from time to time, have to rebalance disks after blade damage has occurred. Yet, this method is simple, quick, and within reach of semi-skilled personnel.

THEORETICAL CONSIDERATIONS.

The Principles.

A perfectly balanced turbine wheel will be in equilibrium in any position if put on a mandril with small trunnions running on carefully levelled and rigidly supported knife edges. If, however, an unbalance weight p is attached to this wheel in position P (see Fig. 1) at H inches above the knife edge plane and at radius R from the trunnion axis, then the disk will swing with slightly damped oscillations until, after a certain time, the weight stops in bottom position. The first full oscillation only is used for measurements. When on the first downward swing, the test weight after leaving position P passes the knife edge plane at P', the kinetic energy of the disk at this moment is :

$$pH = \frac{I\omega^2}{2} \quad .. \quad .. \quad (1)$$

where I is the polar mass moment of inertia of the disk and ω the angular velocity which is inversely proportional to the oscillation period T. Therefore, with β a calculable constant,

$$p = \beta \frac{1}{T^2} \quad .. \quad .. \quad (2)$$

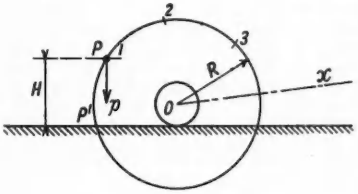


Fig. 1

i.e. the unbalance weight p is inversely proportional to the square of the oscillation period T. If the test weight is now attached to point 2, 3 or any other point of the wheel periphery, this point with the weight then placed in position P and the test repeated, the measured oscillation period T will always have the same value for a perfectly balanced disk. An unbalanced wheel however will show different values, the oscillation period being a minimum (t) when inherent unbalance Π and test unbalance p coincide, and a maximum (T) when p and Π are situated at opposite sectors of the disk. The combined unbalance is in the first case :

$$p + \Pi = \beta \frac{1}{t^2}$$

and in the latter case : $p - \Pi = \beta \frac{1}{T^2}$.

It follows that

$$\Pi = p \frac{\frac{T^2}{t^2} - 1}{\frac{T^2}{t^2} + 1} \quad .. \quad .. \quad (3)$$

This equation can easily be solved in every case by having recourse to nomogram Fig. 2 which gives Π in grams at radius R for a test weight p of 100 grams. When using another test weight, multiply the result by the actual test weight and divide by 100.

Accuracy of Method.

Weight p can be checked accurately. If times T and t are measured to an accuracy of x per cent, then the error in Π is :

$$e_r = \frac{8x\left(\frac{T}{t}\right)^2}{\left(\frac{T}{t}\right)^4 - 1} \text{ per cent.} \quad .. \quad (4)$$

Table I gives the resulting error in Π if $x = 1$ and T/t between 1 and 2, and shows that for T/t approaching unity, the relative error is appreciable but the absolute error in grams remains small and does not appreciably affect the value of this method.

Theoretically, the absolute error in Π would be smallest when using the smallest possible test weight, but practically friction influences increase with small

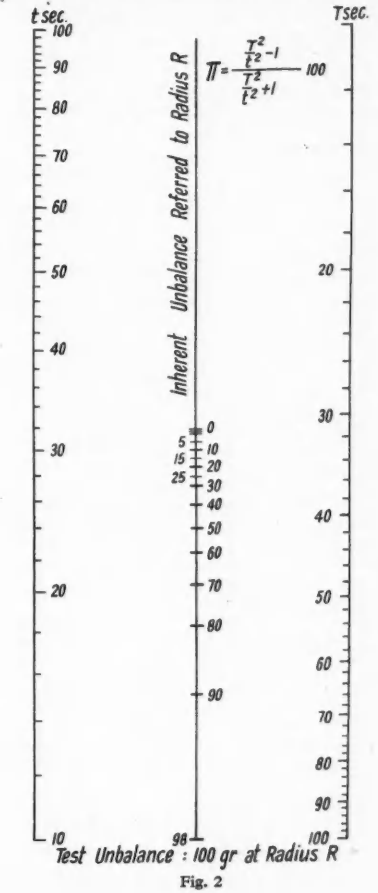


Fig. 2

TABLE I.

$\frac{T}{t}$	e_r per cent	Calculated inherent unbalance for p = 100 gr.	Absolute err r in inherent unbalance in gr.
1·0	—	—	—
1·1	21·1	9·5	2
1·2	10·7	19·6	2·1
1·3	7·27	25·8	1·88
1·4	6·52	32·4	1·79
1·5	4·40	38·5	1·69
1·6	3·69	43·7	1·61
1·7	3·15	48·6	1·53
1·8	2·73	52·9	1·44
1·9	2·40	56·6	1·36
2·0	2·13	60	1·26

oscillation speeds and lead to straying results. Optimum conditions can be calculated and are shown here in the example of Table II. Different test weights (25-100-200-300 grams) were successively attached to the same peripheral point of a perfectly balanced turbine wheel weighing 300 kgs. (660 lbs, PD^2 approx. 4000 lbs. ft.^{2}), and 10 to 20 readings of the oscillation period were taken in every case. Test results (Table II) are closest when using the 100 gram weight, only slightly worse when using heavier weights, but show stray results for the 25 gram weight. The optimum test weight for balancing a disk identical with that used in these tests would therefore be approximately 100 grams heavier than the inherent unbalance, always referred to radius R of the test weight attachment. When balancing different wheels, the initial height H should be maintained and the test weight changed in proportion with the respective mass moments of inertia of the wheels. The measured oscillation periods would thus have similar values, see equ. (1). Incidentally, results shown in Table II allow of checking experimentally the law expressed in equ. (2). Good agreement will be found.

TABLE II.

Test weights							
25 g	100 g		200 g		300 g		
91·3	45·7	46·2	32·4	32·3	26·5	26·6	
93	45·8	46·4	32·4	32·2	26·3	26·8	
93	46·2	45·7	32·3	32·5	26·8	26·5	
96	46·2	45·8	32·6	32·1	26·7	26·7	
90	45·7	46·2	32·6	32·4	26·7	26·8	
90	45·7	46	32·4	32·5	26·6	26·7	
95	46·4	46	32·5	32	26·6	26·8	
91	45·4	45·3	32·4	32·6	26·6	26·7	
95	46·2	45·8	32	32·2	26·9	26·8	
93	45·8	45·9	32·5	32·5	26·7	26·9	
92·75 +3·5% −3%	45·9 + 1·1% — 0·45%		32·4 + 0·6% — 1·25%		26·7 + 0·75% — 1·5%		

THE PRACTICAL APPLICATION.

Test arrangement.

In a power station, the balancing beam supports were erected on the flange of the L.P. turbine casing which provided a perfect base. Rigid steel bars which had served as a base for an alternator housing were available and were used as beams. The knife edges were carefully dressed, polished, clamped on top of the beams, and precisely levelled by spare coupling bolts which had been provided for the supports and acted as screw-jacks. The test weights (100, 200 grams) consisted of weighted spring pincers which could easily be fixed to the blade shrouding and offered several advantages over the traditional plasticine balls. A scribing block was used to check the initial height H above the top flange of the balancing beams, this flange being used as the observation plane. Oscillation periods were measured by chronograph. Finally, a portable grinder was used for removing excess material on the disk.

Test method.

The wheel with its precisely equilibrated mandril is carefully placed on the knife edges and left to swing freely. When in equilibrium, the wheel is marked L at its lowest, and I at its highest point. A test weight chosen in accordance with the rules explained above is

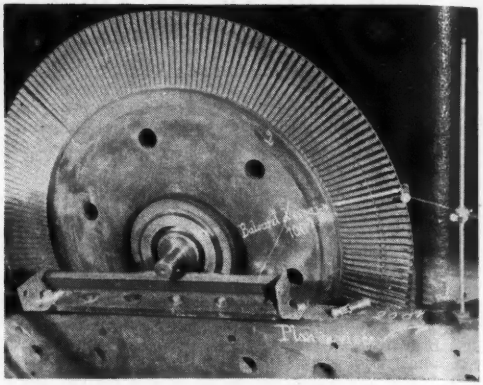


Fig. 3. **Test Arrangement.**

attached to the shrouding at point L. The scriber point is adjusted to approximately 1 foot above the observation plane and the disk turned until a mark on the test weight is level with the point. The disk is then left to swing freely without any initial impulse. At the precise moment the mark on the test weight passes through the observation plane for the first time, the chronograph is started, and it is stopped when the weight, after returning to the highest position of the swing, passes the observation plane for the second time. If the disk does not swing back far enough, the test weight must be increased. The chronograph reading is noted, the weight attached to a point 3 or 4 blades away from the first point and the test repeated. The minimum time t is thus bracketed and accurately measured after a few tests. The point of attachment of the weight when the minimum time is measured, indicates the exact zone of unbalance. If the oscillating time is identical for all these tests, then the wheel is already balanced. A few more tests around point I establish the maximum oscillating time T, and nomogram Fig. 2 then indicates the actual amount of inherent unbalance Π referred to the radius R, where the test weight is attached.

Excess material in the zone of unbalance near point L is then ground away and the operation checked from time to time by noting the time difference $T - t$. The aim is to reduce this difference to zero, and the reduction from initial value to the value when checking is a good indication as to how much material remains to be removed. The check also allows to discover, and follow up by grinding, any shift in the zone of unbalance. When the difference $T - t$ is reduced to a few seconds, check test should be taken at right angles from the points L and I and grinding continued wherever the difference $T - t$ is the larger.

The balance can be considered satisfactory when check tests at eight equally distributed points on the periphery result in oscillating times which deviate from the average reading by not more than 0·5 seconds. According to equ. (3), the remaining unbalance would then be less than 1 gram for a test weight of 100 grams and an average measured oscillating time of approximately 50 seconds.

CONCLUSIONS.

If a wheel cannot be balanced dynamically, it should be statically balanced, as precisely as possible. The method described here offers a quick, easily understandable, accurate and objective method, superior to the old traditional methods, and can be used by any intelligent fitter. It is, however, recommended to take a last check by another method in order to make sure that a neutral equilibrium exists on two diameters at right angles to each other.

SHUTTER-TYPE DUST COLLECTOR OF SMALL DIMENSIONS.

By N. I. Zverev. (From *Izvestya Vsesoyuznogo Teplotechnicheskoga Instituta*, Vol. 15, No. 3, 1946, pp. 12-15, 9 illustrations.)

At the present time the majority of Soviet power stations operate on coarsely ground low grade fuel of high ash content. The dust content of the flue gases is therefore very considerable, and the flue dust contains a large percentage of coarse particles, which leads to rapid wear of the induced draught fans. Thus, for instance, the service life of the induced draught fan rotors at the Krasnogorsk plant amounts to no more than 3 to 4 weeks.

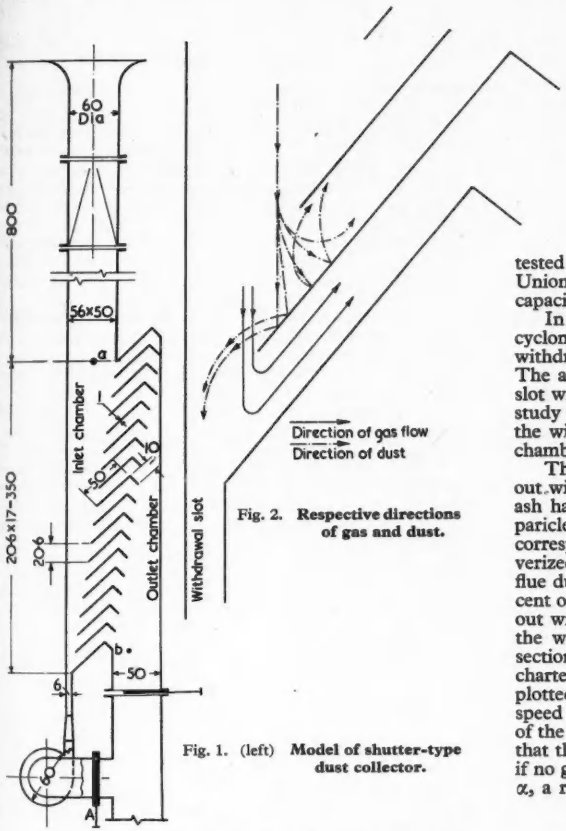


Fig. 2. **Respective directions of gas and dust.**

Fig. 1. (left) **Model of shutter-type dust collector.**

In order to combat the wear of induced draught fans the wearing surfaces are provided with wear-resistant hard metal coatings, special wear resistant rotor constructions are resorted to, and dust collectors are installed ahead of the induced draught fans. It has, however, been found in practice that the first two of these counter-measures do not effect a sufficient reduction of fan wastage. The third counter-measure, that is, the provision of flue dust collectors, constitutes the most radical solution to the problem. At the Kashira plant, for instance, the installation of Davidson collectors of 42 per cent overall efficiency has increased the service life of the rotor vanes of the induced draught fan to 1-1·5 years as compared to a service life of 1-1·5 months in the case of the boilers not equipped with dust collectors

However, the dimensions of the usual dust collector types are so large that they cannot be fitted to existing boiler installations ; also, their manufacture requires a considerable amount of steel. In many cases it is therefore impossible to resort to the installation of dust collectors. In 1943 the All Union Institute undertook the task of developing a dust collector of small dimensions suitable for protecting induced draught fans against wear. This work led to the development of the shutter-type dust collector, as illustrated in Fig. 1. Here the gas entering any one of the upwardly inclined passages between the individual shutters is subjected to a sharp deflection in the direction of flow, with the result that the coarser particles retain their original downward motion, and thus impinge upon the projecting edge of the shutter forming the bottom of the passage concerned (Fig. 2). The retained dust particles finally fall into the withdrawal slot at the bottom of the inlet chamber, from whence they are conveyed through a duct into a small auxiliary cyclone ; the clean gas delivery of the latter is seen to be connected with the clean gas duct of the main collector. During the period 1944-1945 no fewer than 16 various designs of this type of dust collector were tested at the laboratory for gas cleaning of the All Union Institute, the individual models being built for a capacity of 200 cu.m. per hour.

In order to keep the dimensions of the auxiliary cyclone small, the amount of gas discharged into the withdrawal slot should be made as small as possible. The amount of gas discharged through the withdrawal slot will be determined by the gas speed in same, and a study was therefore made of the ratio of the gas speed in the withdrawal slot to that at the entrance of the inlet chamber of the collector.

This and all subsequent investigations were carried out with the use of the fly ash of lean Donetz coal, this ash having a residue of 21 per cent on a sieve passing paricles of 48 micron maximum size. This flue dust corresponds to that obtaining in boilers fixed with pulverized Lower Moscow coal. Thus, for instance, the flue dust at the Kashira plant shows a residue of 18 per cent on the aforementioned sieve. All tests were carried out with a withdrawal slot of 6 mm. width, the area of the withdrawal slot representing 10·7 per cent of the sectional area of the inlet chamber. The test results are charted in Fig. 3, where the collecting efficiency is plotted against the ratio α, that is, the ratio of the gas speed in the withdrawal slot to that in the inlet chamber of the collector. Referring to this graph, it will be seen that the collecting efficiency amounts to only 4 per cent if no gas withdrawal takes place. With rising values for α, a rapid increase in collecting efficiency is registered

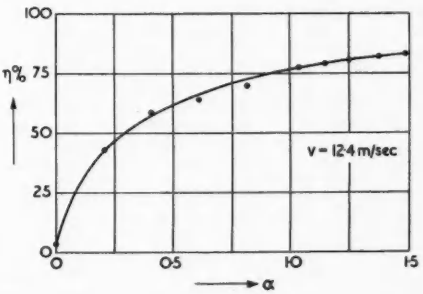


Fig. 3. **Influence of relative gas speed in withdrawal slot upon collecting efficiency.**

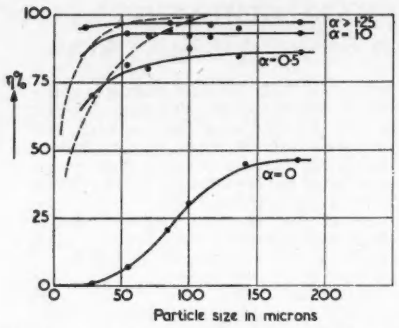


Fig. 4. **Fraction collecting efficiency *versus* relative gas speed in withdrawal slot.**

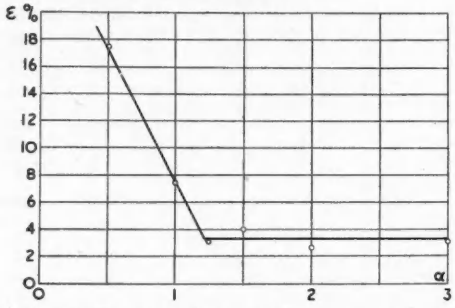


Fig. 5 **Relationships between relative gas speed in withdrawal slot and factor ε in the range of particle sizes from 30 to 420 microns.**

until a value of $\alpha = 1$ is reached. Any further increase in α results only in a slight further improvement in efficiency.

In the case of dust collectors to be used for the protection of induced draught fans, particular importance attaches to the coarse particle collecting efficiency, as it is mainly the coarse fractions which cause wear. For this reason another series of tests was made in which the influence of α upon the fraction efficiency was investigated. As will be seen from Fig. 4, the ratio α exerts an extremely large influence not only upon the overall efficiency, but even more so upon the fraction efficiency, the latter sharply increasing with rising values for α. This graph also includes curves for the fraction efficiency of a Davidson cyclone installed at the Kashira plant and of a cyclone of the multi-unit design.

From a comparison of the various collecting efficiencies it will be seen that in the range of large particle sizes the shutter-type dust collector excels the normal cyclone types and is hardly inferior to the multi-unit cyclone type. The shutter type dust collector therefore fully meets the demands made with regard to collecting efficiency. A number of special tests were run in order to determine the optimum value for α as far as the collection of coarse fractions of 30-420 micron particle size is concerned. The relationship between α and $\epsilon = 100 - \eta$, as found in these tests, is charted in Fig. 5. Up to a value of $\alpha = 1{\cdot}25$ the factor ϵ is seen to decrease sharply, while with larger values for α this factor remains constant. The value of $\alpha = 1{\cdot}25$ therefore represents the optimum for the collection of coarse particle fractions, the corresponding collecting efficiency being $\eta = 96{\cdot}6$ per cent. The fraction efficiencies obtaining with $\alpha \geq 1{\cdot}25$ are also charted in Fig. 4. An efficiency of 96·6 per cent diminishes the wear of the induced draught fan by $100/3{\cdot}4 = 29{\cdot}4$, that is, by some 30 times, as far as this range of particle sizes is

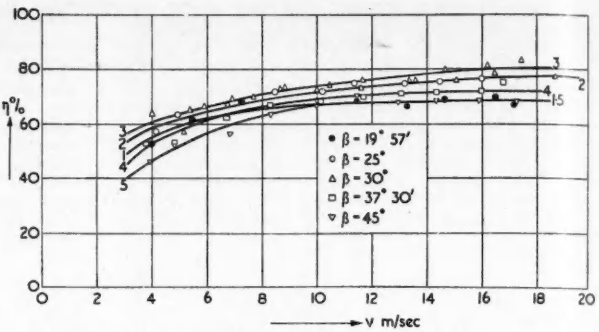


Fig. 6. **Influence of shutter angle and gas velocity upon collecting efficiency.**

concerned. However, the concurrent collection of smaller particles of 0 — 30 microns diameter, although carried out less efficiently, has also a share in reducing wear.

In order to find the optimum angle of incidence β of the shutters, a number of special tests were carried out with the angularities $\beta = 19$ deg. 57′ ; 25 deg. ; 30 deg; 37 deg. 30′, and 45 deg., the pitch of the shutters being 20·6 mm. The test results are charted in Fig. 6, where the collecting efficiency η is plotted against the air velocity prevailing in the inlet chamber of the dust collector. In order to determine the optimum angularity β, the collecting efficiency is also plotted against the angularity β for various air velocities, as shown in Fig. 7. Referring to this chart, it will be seen that optimum overall collecting efficiency obtains with $\beta = 30 \div 31$ deg. The topmost curve included in Fig. 7 refers to the collecting efficiency with regard to a particle size range from 30-420 microns ; and it will be seen that for this range a value of $\beta = 27$-28 deg. yields optimum results. From the combined aspects of overall efficiency and "coarse fraction collecting efficiency," an angularity of $\beta = 30$ deg. can therefore be considered to represent the optimum shutter incidence.

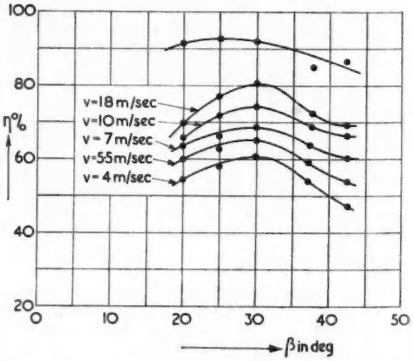


Fig. 7. **Influence of shutter angle upon collecting efficiency for different gas velocities.**

In the course of these tests the influence of the angle β upon the difference in static pressure between the points *a* and *b* (Fig. 1) was also determined. It was found that with increasing values for β the draught loss coefficient continuously decreases, a value of $\xi = 3{\cdot}8$ obtaining with $\beta = 30$ deg. This compares with a value of $\xi = 1645$ for a Davidson cyclone of the Type D-9 ; and the draught loss coefficient of the shutter type dust collector is therefore 70-430 times smaller. For an identical draught loss of the two collector types, the flow area of the shutter-type collector will therefore be

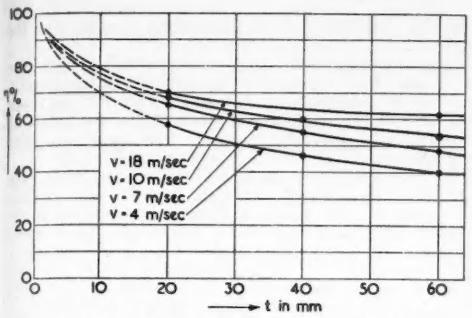


Fig. 8. **Influence of shutter pitch upon collecting efficiency for different gas velocities.**

$1/\sqrt{30}$ to $1/\sqrt{430}$, that is, 8·4 to 20·8 times smaller than that of the centrifugal cyclone. This means that the shutter-type collector fully meets requirements with regard to small dimensions.

All the tests referred to above were conducted on collectors in which the pitch of the shutters was 20·6 mm. For the sake of facilitating manufacture this distance should, however, be of the order of 50 mm. In order to determine the relationship between the shutter pitch and the collecting efficiency, a series of tests was run with shutter pitches of t = 20, 40, and 60 mm., the shutter angle being β = 19 deg. 57′. The test results are charted in Fig. 8, where it will be seen that the collecting efficiency drops with increasing pitch t. However, the collecting efficiency for the coarse particle size range from 30 to 420 microns was found to be unaffected. Therefore the fall in overall efficiency shown is due to a decrease in fraction collecting efficiency for particle fractions below 30 microns.

On the basis of this experimental data and in consideration of theoretical findings, a shutter pitch of t = 50 mm. was adopted, each collector to comprise from 50 to 100 shutters. This results in a collecting efficiency of 50 per cent.

Two commercial installations were made in 1944. Tests conducted on one of these installations yielded an efficiency of 43 per cent with a dust having a residue of 20 per cent on a sieve passing particle sizes up to 48 microns. The small efficiency recorded is due to an abnormally low value for ξ resulting in a value of $\alpha = 0{\cdot}5$, which, as was shown above, must result in a lowered efficiency. With values for α ranging from 1 to 1·25, this plant would show a collecting efficiency of not less than 50 to 55 per cent.

HOLLAND

THE NORMAL CATHODE FALL FOR MOLYBDENUM AND ZIRCONIUM IN THE RARE GASES.

By T. Jurriaanse, F. M. Penning and J. H. A. Moubis. (From *Philips Research Reports*, Vol. 1, No. 3, April, 1946, pp. 225-238, 11 illustrations.)

In this paper a procedure will be described for obtaining a reproducible cathode fall which is constant with time. The value of the normal cathode fall V_n obtained in this way we consider as that of highly purified molybdenum.

The tubes used are shown in Fig. 1. They contain a molybdenum rod anode and a molybdenum plate cathode. The cathode is simply pinched upon the nickel poles of the glass seal and cleaned thoroughly. After sealing in, baking out the tube at high temperature and heating the cathode with high-frequency currents, the gas is admitted. Then a small glow current of, for instance, 20 mA is switched on, until the glow contracts and the voltage decreases. As the cathode fall of nickel is high, the glow always contracts on the molybdenum. This circumstance allows a very simple cathode construction of the tube with unscreened nickel poles. When the glow has contracted, the current is raised as high as necessary for sputtering the cathode material care being taken that the glow never reaches the nickel poles, since sputtered nickel immediately spoils the cathode surface, resulting in an incorrect value of V_n.

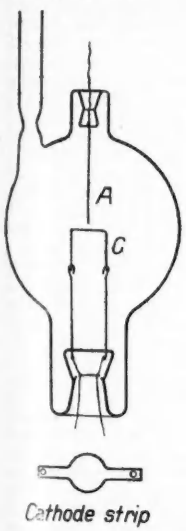


Fig. 1. **Discharge tube with cathode construction.**

The cathode being cleaned and outgassed in this way, the gas is renewed and the tube sealed off from the pump. This operation again spoils the cathode, so that sputtering is again necessary. But now the sputtering has to be continued until the walls of the bulb are covered with a sufficiently thick layer of the sputtered material. *This is an essential requirement for reproducibility and stability*. Without this layer on the wall, it is indeed possible to clean the cathode surface with high current and to obtain for some seconds a reproducible and low value of the cathode fall for different specimens of tubes, but after burning for some time on a low current, the voltage rises and the glow may contract again.

Good results, that is to say, a variation of some tenths of a volt per 1000 burning hours with the normal current density (see Fig. 2) were obtained by covering the tube walls heavily. This sputtering may be effected in a short time at a high current, raising the temperature up to thermal electron emission temperatures.

Higher temperatures are not desired, as the voltage then drops again and the discharge may concentrate on one point of the cathode, melting the molybdenum. It is also possible, however, to sputter for a long time at a relatively low current density. We found for various tubes with cathodes of 2cm² and filled with 20 mm neon, sputtered with currents between 25 mA and 350 mA, the same normal cathode voltage of 110V to 112V and voltage variations per 1000 burning hours of about 0·1

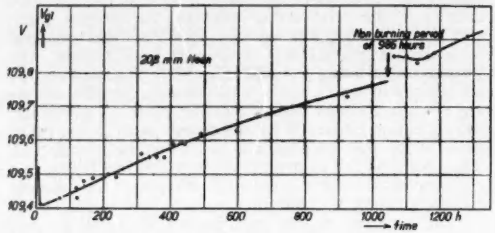


Fig. 2. Glow voltage of a molybdenum cathode in neon as a function of time.

volt. The light absorbing power of the walls was held the same for these tubes.

From this result we may conclude that a high temperature of the cathode during sputtering is not necessary for obtaining reproducible values of V_n. The characteristic of the abnormal glow shows extremely low voltages, as can be seen from Fig. 3 for the gases argon and neon. The maximum shows that at higher currents the flow gradually changes into a thermionic arc, the temperatures measured in the argon discharge (Fig. 3) being in agreement with this conclusion.

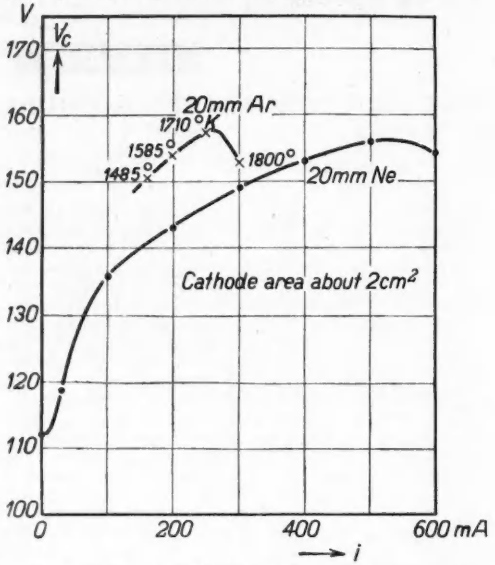


Fig. 3. i—V characteristics of molybdenum at high currents, in neon and argon.

The highest possible current in argon for the glow discharge is smaller than in neon, because the heat conductivity is smaller in the former case.

The thickness of the cathode material is of importance in so far as a thin material more easily leads to arcing instead of glowing, and thick material is difficult with respect to outgassing. We obtained good results with molybdenum plate of 0·2 to 0·3 mm.

It was mentioned above that the sputtered layer on the wall is essential for the reproducibility and stability of the normal cathode fall. This is proved by the circumstance that valves with clear or very slightly covered walls always had an unstable cathode fall. The function of this layer can be explained in different ways, e.g.

(a) It is possible that the molybdenum layer acts as a getter absorbing the gases resulting from the last action of the pumping process, e.g. sealing off. Consequently the layer must not necessarily be deposited on the wall but may also be introduced elsewhere in the tube.

(b) Essentially the layer must be deposited on the wall, absorbing the gas coming from the walls directly during the time the tube is in use and before it can reach the cathode. Therefore every part of the wall must be covered with sputtered material. In these two cases the function of the layer is physically the same.

(c) The sputtered layer acts as a screen, preventing the liberation of gases from the walls by the electric discharge. It is known, for example, that neon in the metastable state liberates oxygen from glass parts of the tube.

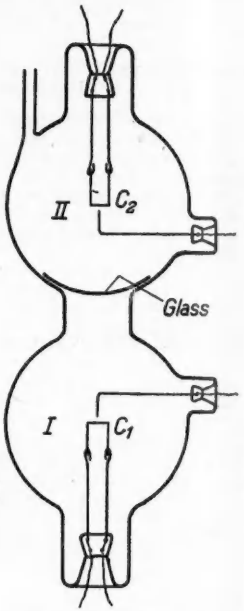


Fig. 4. Double discharge tube for studying the influence of a molybdenum layer on the glass wall.

A combination of (a), (b), and (c) is, of course, also possible, and we shall prove presently that the sputtered layer shows all these functions. At first we shall show that it is indeed necessary to cover the whole of the wall facing the cathode with sputtered material, the presence of a sputtered layer in another part of the tube not being sufficient. The tube used (Fig. 4) consists of two sections, section I containing the measuring cathode C_1. In section II a getter layer can be obtained by sputtering the auxiliary cathode C_2. At first the cathode C_1 was prepared by a heavy glow but during such a short time that no perceptible layer was obtained on the wall. A low voltage of V_n was reached, but it increased with time as shown in Fig. 5, due to contamination. Then C_2 was sputtered heavily with the little glass plate in a position to separate the two sections, the sputtering material remaining only in the upper section. When the measuring current of C_1 was switched on again, the two sections being in contact with each other, the voltage was somewhat higher at first, showing possibly the clean-up of contaminations, as the cathode fall in pure neon is higher than in impure neon. But soon the voltage dropped, the cathode being cleaned under the influence of the measuring current. After burning 140 hours a minimum was reached and the voltage slowly rose again. The gas and the cathode were contaminated anew, and the cathode of the upper section had to be sputtered once more for a considerable time in order to clean the gas and the lower cathode again. It was only after sputtering the lower cathode to such an

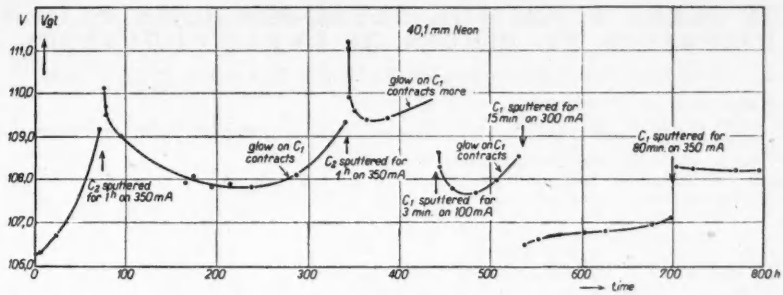


Fig. 5. Variation of V_{gl} due to the sputtering of the cathodes C_1 and C_2 of the double discharge tube shown in Fig. 4.

extent that the wall was firmly covered with molybdenum that the voltage had a stable and low value for hundreds of hours. We may therefore conclude from this experiment that molybdenum acts as a getter for traces of impurities, but that it is not sufficient to introduce the molybdenum layer in a remote part of the tube.

We found that if the tubes were allowed to rest and the glow voltage was measured before and after the resting period, its value remained constant when every glass part was covered with the molybdenum layer. But when large parts of the wall were left uncovered or if the construction of the tube was such as to leave large dead ends not covered with molybdenum, the cathode was more and more completely spoiled, the longer the resting period was taken. Obviously there are always little gas residues coming free from the walls and spoiling the cathode surface if they are not removed immediately. This is achieved when every part of the glass wall is covered with its own getter.

VALUES OF THE NORMAL CATHODE FALL FOR MOLYBDENUM IN THE RARE GASES AND SOME GAS MIXTURE.

The value of the normal cathode fall may be measured in several ways, all giving about the same results. One of the easiest ways is to measure the voltage-current characteristic between parallel plates at a suitable distance and taking the minimum voltage which is reached. At higher pressures this minimum voltage is found over a large range of currents. Our experiments were made with tubes of the type shown in Fig. 1, in order to allow the sputtered material to reach the whole of the glass wall. Nevertheless, also with these tubes a considerable part of the characteristic showed the flat region mentioned above, and the minimum voltage in this region may be considered approximately as representing the value of the normal cathode fall. In all cases the cathode-anode distance was chosen so small that no anode fall could be developed.

The results are shown in Table I and in Fig. 6. For each gas the number of tubes used is mentioned, also the gas pressure, the mean value of V_n and the extreme values.

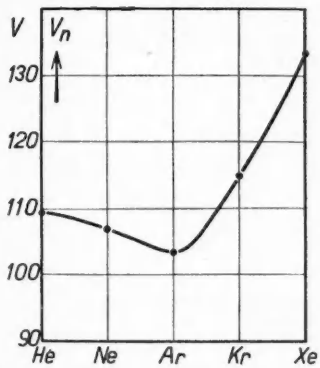


Fig. 6. The value of the normal cathode fall V_n of molybdenum in rare gases.

TABLE I.

Gas	Pressure in mm	Number of tubes	V_n^m	Extr V_n	
				low	high
neon	40	16	107·1	105·5	108·9
neon	20	8	110·6	109·4	112·6
argon	20	5	103·4	102·3	104·5
argon	10	5	103·6	102·6	105·0
helium	42	1	109·5	—	—
krypton	13	1	114·8	—	—
krypton	7·1	1	119·8	—	—
krypton	5·8	1	120·2	—	—
xenon	13	1	133·6	—	—
xenon	5·4	1	134·4	—	—
$Ne+\frac{1}{2}\%Ar$	40	4	85·0	84·3	85·8
$He+\frac{1}{2}\%Ar$	17·5	1	106·3	—	—

The table shows that the results for V_n are reproducible within $\pm$ 1 or 2 volts in so far as values for the same pressure are considered. As a matter of fact we always obtained lower values at higher gas pressures.

THE NORMAL CATHODE FALL FOR ZIRCONIUM.

For zirconium it is sufficient to give a table of data (Table II) because the phenomena are almost the same as with molybdenum. Only the preparation of the cathode is more difficult and must be done carefully because of the lower melting point of this material, so that the time necessary for blackening the glass walls is longer than with molybdenum. But even with the greatest precautions that could be taken, we could not obtain a stable voltage in the gases krypton, xenon and $Ne + \frac{1}{2}$ per cent Ar, the low voltage immediately rising after sputtering the cathode heavily.

TABLE II

Tube	Gas	Pressure in mm	V_n
J.D. 28	neon	42·7	103·7
J.D. 45	neon	39·5	105·7
J.D. 351	neon	20·4	105·1
J.D. 35 No. 1	neon	19·5	108·0
J.D. 45 No. 2	argon	10·2	92·1
J.D. 45 No. 3	argon	7·1	92·2

OPTIMUM BLADE PITCH FOR TURBO-MACHINES WITH SPECIAL REFERENCE TO BLADES OF GREAT CURVATURE.

By O. ZWEIFEL. (From *Brown Boveri Mitteilungen*, Vol. 32, No. 12, December, 1945, pp. 436-444, 11 illustrations.)

No generalized statement on the optimum spacing of blades can be found in the steam turbine literature known to the author.

The lift coefficient of the aerofoil theory as used in axial-flow blower calculation[1,2], varies with the angle of flow to such an extent that, at large deflections of the flow, it is not possible to estimate its magnitude. That is to say, there is no theory available for the optimum blade pitch, i.e. the maximum loading per blade.

Based on the author's work, the *Brown Boveri Co.* has, since 1936, been using a coefficient of aerodynamic loading Ψ that proved to be a useful criterion in the calculation of pitch for cascades of blades.

NOTATIONS

		Reference: Figure	Equation
u	Peripheral speed	9	—
c	Absolute velocity of flow	9	—
w	Relative " " "	9	—
Δw_t	See Fig.	5	—
β	Angle of relative velocity	9	—
Γ	Circulation	—	2
p	Static pressure	4	—
p_{ges}	Total "	4	—
A	Lift per unit length of blade	1	1
N	Normal force	1	—
T	Tangential force	1	12
M	Torque acting on blading	—	22
C	Coefficient of loading referred to w_∞	—	5
ψ	Coefficient of loading referred to w_2	—	13
ϵ	Drag-lift ratio	—	19
$b\ s\ s'\ t$	See Fig.	1	—
$a\ r$	" "	2	—
$h\ R$	" "	9	—
S	Static moment $\int R.df$ of blade area turned into meridian plane	—	—
z	Number of blades	—	—
ρ	Density of medium $= \left(\frac{\gamma}{g}\right)$	—	—

	INDICES:	See Fig.
.. 1	Blade inlet	9
.. 2	Blade discharge	9
.. ∞	Flow without circulation	1
.. t	Tangential direction	9
.. n	Perpendicular direction to peripheral velocity (meridian component of velocity)	9
.. A	Lift	1
.. T	Tangential force	1
.. W	Drag	–
.. A1	Lift of single aerofoil	–
.. id	Ideal pressure distribution according to section II.	–

I. THE LIFT COEFFICIENT OF THE AEROFOIL THEORY APPLIED TO ROWS OF BLADES WITH GREAT CURVATURE.

Referring to two-dimensional flow through cascades of incompressible and frictionless media, Fig. 1 shows a row of blades causing great deflection of the flow. w_∞

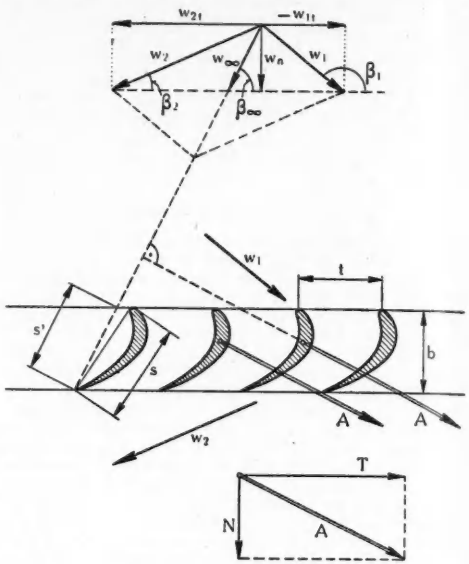


Fig. 1. Velocities and forces with ideal flow through a cascade of blades.

is the ideal speed of parallel flow which remains if the blade-bound circulation is disregarded. Contrary to the single aerofoil section, the flow pattern of the infinitely long blade row is changed, up to infinity, both in front of and behind the row. The lift A acting on a single blade is perpendicular to w_∞, and its magnitude per unit length of blade is :

$$A = \rho \cdot w_\infty \ (w_{2t} - w_{1t}) \cdot t \qquad .. \qquad (1)$$

as can easily be derived from the impulse and energy equations. In this equation we have for the circulation Γ of one blade in the cascade :

$$\Gamma = (w_{2t} - w_{1t}) \cdot t \qquad .. \qquad .. \qquad (2)$$

$$A = \rho \cdot w_\infty \cdot \Gamma \qquad .. \qquad .. \qquad (3)$$

A further relationship which will later be necessary can be seen from Fig. 1.

$$\frac{T}{A} = \frac{w_n}{w_\infty} = \frac{b}{s'} = \sin \beta_\infty \qquad .. \qquad (4)$$

The lift coefficient C_A for the single aerofoil section is defined by :

$$C_A = \frac{A}{\frac{\rho}{2} w_\infty^2 \cdot s} \qquad .. \qquad .. \qquad (5)$$

where s is the chord of the section.

This definition can be applied direct to the blade row by combining eq. (1) and eq. (5). Then

$$C_A = 2 \cdot \frac{w_{2t} - w_{1t}}{w_\infty} \cdot \frac{t}{s} \qquad .. \qquad .. \qquad (6)$$

(1) BAUERSFELD: Z. V.D.I, Vol. 66, 1922, p. 461 & 514; and (2) KELLER: "*Axial-gebläse vom Standpunkt der Tragflügeltheorie*", Dissertation, Zürich, 1934.

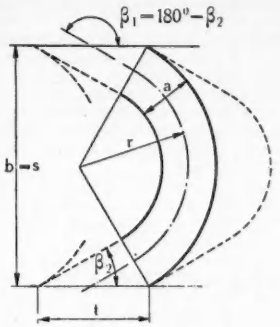


Fig. 2. **Impulse cascade with identical inlet and outlet angles.**

or, expressed by the angles

$$C_A = 2 \sin \beta_\infty \, (\text{cotan } \beta_2 - \text{cotan } \beta_1) \cdot \frac{t}{s} \quad .. \quad (7)$$

Eq. (7) would be a useful relation for determining the best pitch if the optimum lift C_A varied but slightly with the flow angles; that this is not the case can be shown on hand of an example. Fig. 2 shows the passage between two blades of an impulse row, so chosen that the width of the passage and its curvature remain constant with a view to obtaining a simple relationship. From pure geometry we obtain :

$$\frac{s}{t} = \frac{b}{t} = \left(\frac{r}{a} + \frac{1}{2}\right) \sin 2\beta_2 \quad .. \quad (8)$$

For thin sheet-metal blades this expression is reduced to :

$$\frac{s}{t} = \frac{b}{t} = \frac{r}{a} \sin 2\beta_2 \quad .. \quad .. \quad (8a)$$

According to Briling[3] optimum conditions are obtaining for $a = \frac{1}{2} r$ (also an optimum condition for pipe bends).

Hence :

$$\frac{s}{t} = \frac{b}{t} = 2{\cdot}5 \sin 2\beta_2 \quad .. \quad (9)$$

or for sheet metal blades :

$$\frac{s}{t} = \frac{b}{t} = 2 \sin 2\beta_2 \quad .. \quad .. \quad (9a)$$

In Fig. 3 three blade rows are selected with β_2 values 30 degrees apart, all satisfying the optimum condition of eq. (9). The three rows are of the same width, and it may be noted that for both extremes of flow deflection a large pitch is obtained. Introducing eq. (9) into eq. (7) and bearing in mind that for impulse blading cotan $\beta_1 = -$ cotan β_2 and sin $\beta_\infty = 1$, the optimum lift coefficient becomes :

$$C_{A\,opt} = \frac{0{\cdot}8}{\sin^2 \beta_2} \quad .. \quad .. \quad (10)$$

(3) Forschungs-Arbeit H. 68. See also Stodola "*Dampf und Gasturbinen*", 6th. Ed., p. 148.

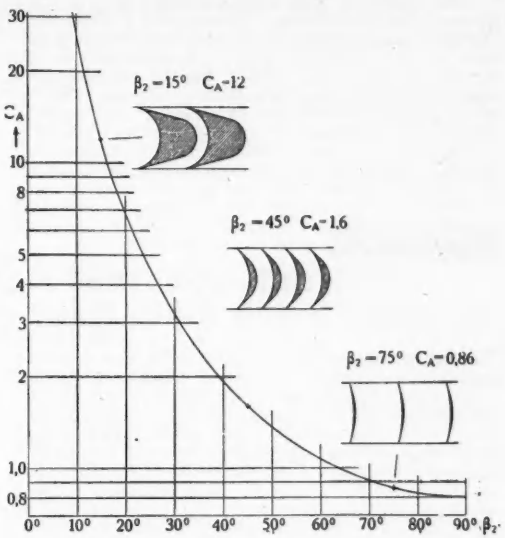


Fig. 3. **Lift coefficient C_A for impulse cascades of optimum pitch according to Briling.**

This function is plotted in Fig. 3. While for small flow deflections the lift coefficient converges toward $C_{A\,opt} = 0{\cdot}8$ thus confirming the optimum values for aircraft wings, it increases without limit for large deflections. From this fact it can be concluded that the lift coefficient, as defined by eq. (5), is, due to its rapid variation with β_2, unsuitable as a criterion in calculating the optimum pitch of blade rows.

II. THE COEFFICIENT OF AERODYNAMIC LOADING.

In this chapter it is proposed to show that a new coefficient, the coefficient of aerodynamic loading Ψ, can be introduced which varies but little with the angles of flow and degree of deflection.

Consider the pressure distribution in the turbine blading (Fig. 4). Curve 1 shows the pressure distribution over the front of the blade, curve 2 that over the back. Then, the enclosed shaded area in Fig. 4, represents the tangential force T. The maximum pressure p_{ges} is obtained at the stagnation point; p_1 is the static pressure at blade inlet, p_2 the static pressure at blade outlet.

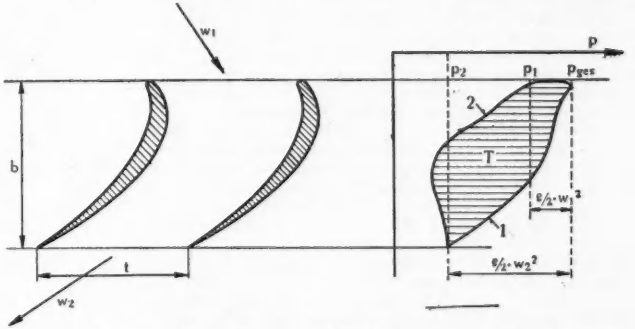


Fig. 4. **Pressure distribution over blades of an accelerating cascade.** 1 .... front of blade. 2 ... back of blade.

The smaller the pitch, the smaller the area T, and the greater the frictional losses. Theoretically it is possible, with increasing pitch, to obtain very low pressures at the back of the blade (increased T) ; however, due to the danger of breakaway, only a limited pressure rise towards the trailing edge is possible in practice. That is to say, a limit is set to the aerodynamic load-carrying capacity of the blade.

To obtain a measure for this capacity, the actual pressure distribution must be compared with the ideal pressure distribution at which the area T is at a maximum without risk of breakaway. Such a distribution of pressure requires a maximum pressure p_{ges} along the whole front of the blade with an abrupt expansion to p_2 at the trailing edge. Similarly, at the back of the blade a sudden expansion from the stagnation pressure p_{ges} to the static back pressure p_2 would have to take place in order to maintain the lowest possible pressure along the back of the blade without pressure rise.

(*To be continued*)

FRANCE

THE ROTATING ELECTRIC ARC FURNACE.

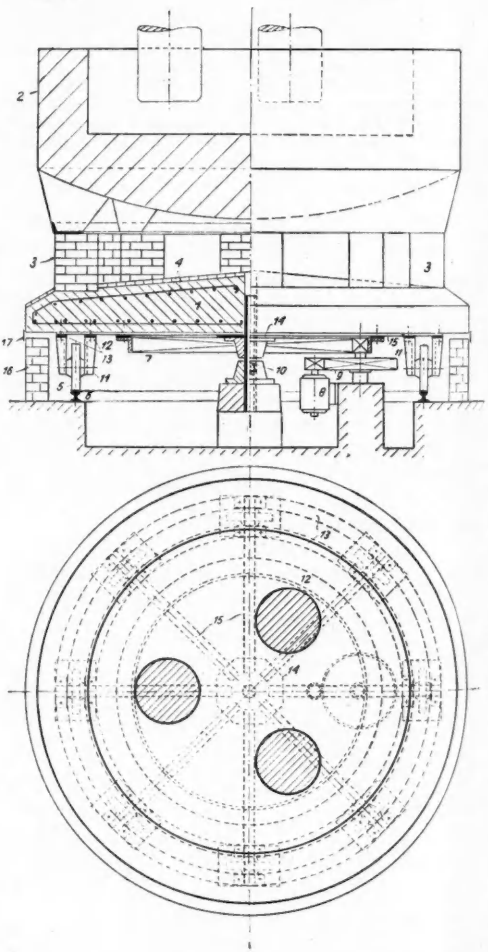
By T. ELLEFSEN. (From *Journal Du Four Electrique et des Industries Electrochimiques*, Vol. 55, No. 3, June/July, 1946, pp. 41-44, 7 illustrations.)

THE functioning of the Electric Arc Furnace is often subjected to " crater " formation which restricts the active zone of the furnace to the immediate vicinity of the electrodes thus leaving a wide " dead zone " which does not take part in the metal fusion. Violent gas eruptions follow and lead to heat losses and wear of the electrode equipment. The active charge is molten too quickly and humidity insufficiently segregated. This crater formation is obviated by the arrangement of a rotating furnace with stationary electrodes (system Elkem-Ellefsen) as erected at Fiskaa Verk (7000 kW) and Porsgrund (6000 kW) in Norway.

The cylindrical furnace vessel is built on a turn-table with 8 wheels running on a rail and is driven by a small electromotor with worm and spur gear reduction of about 1 in 5 million. The rail is levelled first, then fixed to the floor and machined by a planing jig fitted to the centre pivot of the turn-table. The wheels are placed on the rails and fitted with an elastic attachment on to which the turn-table is cast in ferro-concrete. During setting, the turn-table weight is thus equally distributed over the wheels. In order to obtain strictly vertical loading, the rail diameter is held smaller than the furnace diameter, and rail and wheels are protected by a brick wall reaching nearly to the turn-table. The Porsgrund furnace weighs 275 tons, carries 3 electrodes of 30 to 34 inches diameter and its rotating speed can be adjusted from one full turn in 6 hours to one in 96 hours. Actually, a swing of 120 degrees is sufficient to bring the whole charge into contact with the electrodes ; should it be more convenient, the furnace can be made to swing 60 degrees to both sides instead of making one complete revolution. In any case, the mechanism should be reversible so as to allow rotation in either direction. Three tapping holes are provided and a teeming bridge follows the furnace rotation. The Fiskaa Verk furnace has now operated since 1938 without any repairs to the rotating mechanism.

The optimum rotation velocity depends on furnace design, power supply, electrode diameter and distance, composition and bulkiness of the charge. It must be much smaller than the vertical movement of the charge during melting, otherwise the solid top charge would exert undue side pressure on the electrodes.

Tests conducted over 6 months with a stationary and a rotating furnace, both run on FeSi at 75 per cent with the same charge material, showed that the rotating furnace increased production by 17 per cent and saved 10 per cent charge material. Installation costs of the rotating furnace were only 8 per cent higher than those of the stationary furnace. Maintenance and repair costs of the rotating furnace are smaller as wall and floor linings of the vessel and the electrode equipment last longer. The entire charge is always molten and the furnace can be emptied and cleaned very easily when production is interrupted or changed over to another material. Up to now, the rotating furnace has only been used for iron alloys but it also has great possibilities for other materials like calcium carbide. A closed rotating furnace would be specially suitable for smelting in a controlled atmosphere as the formation of cavities and the ensuing irruptions and over-heating would be avoided.



HIGH-SPEED AND SUPER-SPEED CUTTING OF METALS.

By I. M. Besprozvany, A. N. Danielian, A. V. Pankin and N. I. Reznikov. (From *Vestnik Inzhenerov i Tehnikov*, No. 2, 1946, pp. 65-73, 21 illustrations.)

As the quality of cutting alloys improves and machining operations tend towards finer cuts and feeds, metal research workers are beginning to consider cutting speeds running into many hundreds and even thousands of feet per minute.

This tendency received its first open expression in a patent by Dr. Salomon in 1931, who pointed out that at far higher speeds than those at present in use, there are favourable cutting conditions, as indicated in Fig. 1. The speed V_1 is the ordinary cutting speed. V_2, on the other hand, would be attained at a speed which for cast iron would be 15,000 ft./min. However, these " super-speeds " were not obtained in practice and the patent never received practical confirmation.

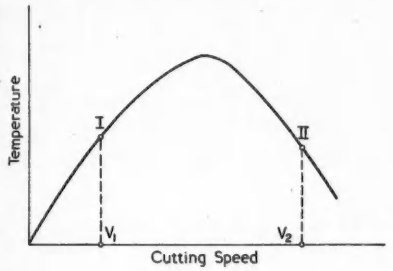


Fig. 1

A practical effort in the same direction was carried out by Shvetsov using a milling cutter like a saw to cut various metals at speeds of 4000 to 6500 ft./min. This resulted in extreme local overheating of the metal by friction at the point of contact of the cutter with the work so that the swarf was actually molten. Thus, the process was not so much one of cutting the metal as of melting it to a depth of some 0·02 to 0·04 in. One of the drawbacks of this method, therefore, was the hardening of the newly " machined " surface.

An alternative approach to the problem of higher cutting speeds was to reduce the resistance of metal to deformation by pre-heating it. Various experiments on carbon steel show that cutting resistance drops rapidly as temperature increases and a method has been developed in which resistance was reduced to one quarter of normal. However, the difficulties with structural changes in the work, the overheating of tool tips and, above all, the inconvenience of manipulating hot metal render this approach quite impractical.

A more fruitful line of research was in the utilization of the heat of cutting, in moderation, to overcome resistance to plastic deformation. Thus Shchelkonogov obtained the following results when working on hardened steel :—

VARYING SPEED.

(" Top rake " angle – 5 deg cut ·01 in., feed ·01 in., Steel " 5 ").*

Velocity (ft./min.)	Tool remained serviceable	Resulting surface
21	5·5 min.	Rough
130	8·0 min.	Poor
700	15·0 min.	Very smooth

* A Russian Standard specification.

VARYING " TOP RAKE " ANGLE OF TOOL.

(Velocity 700 ft./min., cut ·01 in., feed ·01 in., Steel " 5 ").

Top Rake	Tool remained serviceable	Resulting Surface
0	10 min.	Smooth
–5 deg.	15 min.	Very smooth
–10 deg.	13 min.	Smooth

Both the above experiments were carried out without cooling. When a coolant was used the life of the tool was drastically reduced owing to lower temperature of metal at the moment of deformation.

In point of fact, the basis of all super-speed cutting operations lies in softening the metal at the point of contact by heating it, and the amount of this heating depends on the work expended in the cutting operation.

The work of cutting may be subdivided into :—

(1) Work of elastic deformation.
(2) Work of plastic deformation.
(3) Friction of the chip against top of tool.
(4) Friction of tool end against the finished surface.
(5) Shearing of metal in the chip.

Of these, (2), (3) and (4) are predominant, the remainder amounting to only 1·5 per cent. The relative importance of (2), (3) and (4) varies greatly according to the speed of the cutting operation, and a typical diagram, Fig. 2, illustrates how the total work (100 per cent) is subdivided between these items. The diagram refers to carbon steel of 32 ton/sq. in. ultimate, with a depth of cut of ·08 in. and a feed of ·018 in. In this case a speed of 140 ft./min. gives a minimum proportion of frictional resistance and a speed of 355 ft./min. the minimum for work of plastic deformation.

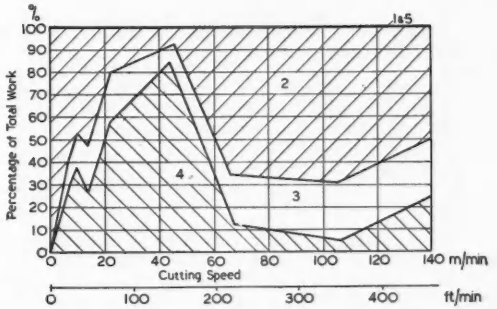


Fig. 2

The figures will vary, of course, for different materials and conditions.

The reason for these changes and for variations in the quality of the finish is to be found in the following phenomena :—

1. For all practical purposes the temperature rises steadily as the speed increases, due to the fact that, at slow speeds, a great deal of heat is led away, while, at higher speeds, more heat is generated per second with no corresponding increase in facilities for conduction. Temperature curves constructed for a point in the metal ·01 in. above the cutting edge do indeed indicate that a maximum is attained at somewhat over 1300 ft/min., but then the metal (steel) has reached about 700 deg. C.
2. Rising temperatures soften the metal.
3. The metal work-hardens due to the work of plastic deformation to which it is subjected.
4. The work-hardening effects may or may not have time to wear off during the process of cutting according to the interplay of temperature and the time available for the " annealing " process.
5. The tool wears differently according to speed and temperature, and the nature of its wear in turn affects the cutting temperature: wear on the top surface

reduces the cutting effort while that of the under surface increases it.

6. Adhesion of metal to the tip of tool, which spoils the finish, reaches a maximum value for different tool angles at between 3 and 50 ft./min. and then reduces rapidly.

7. Degree of surface-hardening of the work—also contributory to poor finish—drops steadily as the speed increases.

As a result of all these factors it is found that speeding up the work with a view to raising metal temperature results in :—

(a) Reduction in the work of cutting by weakening the metal.

(b) Improvement of finished surface.

This has led to a search for heat-resisting cutting materials which would permit of working in temperature ranges where the metal is appreciably weakened.

The use of new hard cutting alloys which can withstand temperatures of 800-900 deg. C. has made it possible to raise cutting speeds in some cases to many hundreds of feet per minute. But these materials have one very appreciable drawback : they are brittle and cannot, therefore, stand up to much vibration and impact. Thus, with cast iron and bronze, which do not call for much cutting effort, these alloys will permit of 3 to 3·5 times the speeds adopted with high-speed steels. But when working on steel, the gain is far from this figure, because the tool tips tend to crumble away.

The latest cutting alloys with increasing percentages of cobalt, wolfram and titanium go some way towards mitigating the position as regards the cutting of steel, but the results are still well behind those which have been attained for cast iron. Moreover, the high temperature leads to excessive wear, even with the hardest of tips.

The problem of wear has been combated by new tool shapes designed to strengthen the tool, reduce abrasion and improve the flow of heat away from the tip.

Increasing the " top-rake " of the tool leads to a big reduction in the work of plastic deformation and so to a reduction both in temperature and in the pressure on the tool.

Thus, in changing the top rake from 5 to 35 degrees with identical feeds, speeds, etc., one authority found that the temperature dropped from 652 to 218 deg. C, and from a curve given by Ernst it appears that the cutting effort would be exactly halved by such a change of angles. At the same time, the quality of the work is greatly improved. Prof. Bezprozvanny found that surface irregularities decreased as follows :—

Top rake angle	20 deg.	25 deg.	45 deg.
Height of ridges	·0070 in.	·0032 in.	·0012 in.

Therefore, where high speed steel is used, a big top rake is usual, but with tools tipped with brittle hard alloys which can withstand high temperatures, it is necessary to make the tip blunt and the rake negative, especially since this moves the center of pressure of the chip against the tool well away from the cutting edge (see Fig. 3).

Negative angles of rake result in greatly increased cutting effort and temperature but they do give the tool tip a reasonable life, —12 deg. being about the best from that point of view, according to Armitage and Smith.

To combine the advantages of a blunt end with reduced chip deformation, tools having a double angle are now being introduced, as shown at A and C in Fig. 4. A tool with a single negative angle is shown at E and one with positive angle at G. The various ways these four types of tool will wear are shown at B, D, F and H respectively. Armitage and Smith recommend —12 and +30 degrees as suitable double angles for steels below a Brinell of 200, and —12 and +15 degrees

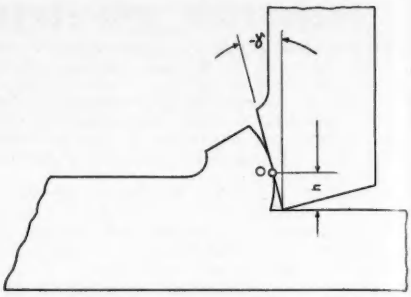


Fig. 3

above that. These angles have been tested by Russian technical authorities and are now officially recommended, but single angles varying from —5 to —10 degrees cannot yet be regarded as definitely superseded and further research work in this sphere is needed.

High cutting speeds have found their best application in milling, particularly end or face milling. But the theoretical suppositions that speeds of several thousand feet per minute will be possible, based on consideration of plastic deformation alone, of which

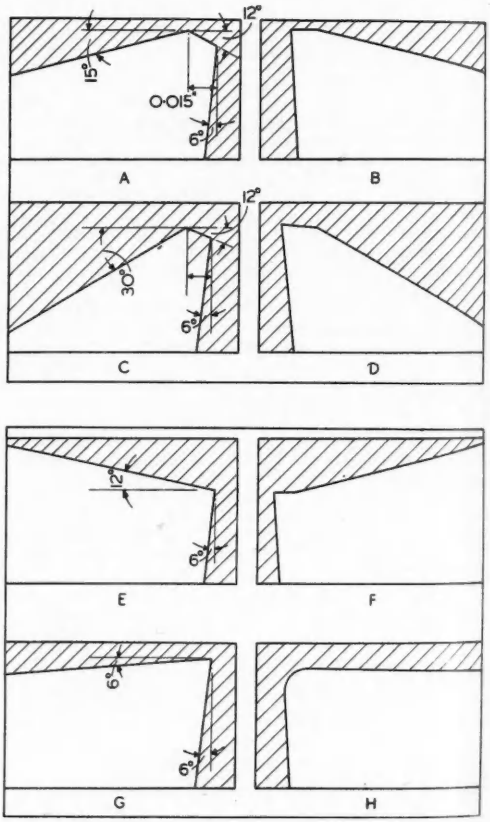


Fig. 4

there has been a great deal of talk in America around 1944, have proved quite wrong in practice. At 3000-6000 ft./min. the metal removed simply burns and the tool life is negligible. The reason for this appears to lie in the fact that friction supersedes plastic deformation as the main heat-producing factor.

As a result, speeds of several thousand feet per minute are for the moment out of the question, except in the case of aluminium with which over 5000 ft./min. has been recorded. As far as steel is concerned, considerations of tool life limit the speed to between 500 and 800 ft./min. and most people appear to prefer speeds between 250 and 500 with a correspondingly heavier cut. In point of fact, therefore, the problem for the moment is simply that of fully utilizing the three-fold increase of speed which the high-speed alloys theoretically justify.

In turning, the difficulties are greater, because the tool is in continuous operation and the cuts are heavier. The speeds obtainable are 330 to 380 ft./min.

PRACTICAL DETAILS OF SUPER-SPEED END AND FACE MILLING OF STEEL.

1. The inclination of cutters, when looking at the side of the tool, must be contrary to that of normal high-speed practice so that the corner where the two cutting edges meet should not be the first point on the tooth to come in contact with the work.

2. As the above will tend to bring the chip downward and choke the space between the teeth, the number of teeth should be kept small.

3. Width of cut should not exceed 0·6 to 0·8 of the tool diameter.

4. Tooth angle in plan view 60-75 degrees.

5. Front relief angle 12-14 degrees.

6. With heavy cuts (0·2 in.) speeds to be 300-500 ft./min.

7. Feed per tooth with most cutting alloys is best limited to ·004 in., but with the toughest of them ·008 to ·012 in. or even ·015 in. are possible.

8. Depth of cut for mild steel and other steels in soft conditions to be 0·2 to 0·5 in. For hard or tough steels 0·1 in.

9. No cooling, as it results in cracked tool tips.

10. American feed (i.e. feed in the same direction as the rotation of tool) is advantageous because the first contact takes place away from the edge.

11. In choosing the type of cutting alloy, toughness is rather more important than hardness, since there is more trouble with crumbling than wear.

12. Wear of teeth is best measured on the back of the tooth and re-grinding should take place when wear marks reach a length of 0·06 to 0·08 in.

13. Hard alloy strips can be either mounted on removable teeth or direct on to the body of the tool. The former method gives a better chance of adjustment and replacement, the latter gives better rigidity.

14. In adopting an existing machine to super-speed work, the number of teeth can be determined from the formula given by Lucht :—

$$\text{No. of teeth} = \frac{0{\cdot}56 \times \text{h.p.}}{t\, S_z\, B\, n}$$

where t = depth of cut, S_z = feed per tooth, B = width of cut, n = r.p.m. (all dimensions being in decimetres).

15. If the h.p. of the machine is so low as to permit of only one tooth working at the desired speed, a single tooth arrangement is quite practicable and effective. The feed can be somewhat greater than the feed per tooth with multiple cutters.

16. High inertia effect is essential, so that the tool must be massive and if necessary a flywheel added. With cutters below 8 in. diameter flywheels are definitely advisable.

17. Great care must be taken when grinding the tools to avoid overheating or initiating cracks. Very fine finish is required.

18. Diameter of the tool must be true to within about ·001 to ·0015 in. otherwise the more protruding teeth will take heavier cuts and soon show sign of wear.

19. The whole question of super-speeds depends largely on the cost of re-sharpening tools. To reduce it, cutting speeds have to be kept down, and the study of sharpening methods is therefore of primary importance.

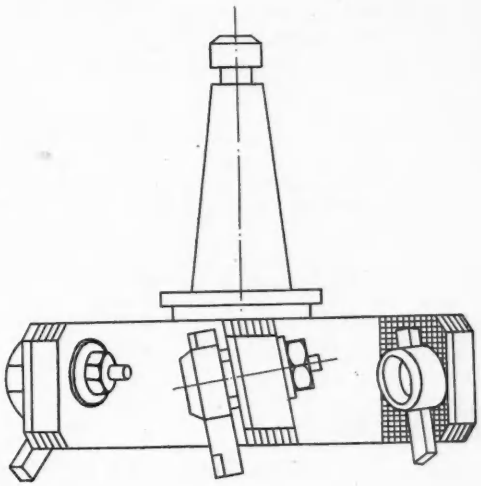


Fig. 5

20. Fig. 5 indicates a universal face-milling head especially designed for super-speed work by the Russian Hard Alloys Technical Bureau. The tools are sharpened individually and set in position to gauge. This reduces sharpening time appreciably, permits of individual replacements and is very accurate.

MILLING MACHINE DESIGN.

Although very good results are obtainable at super-speeds, even with existing machinery, the introduction of super-speed milling really requires entirely new machines with about three times the present normal horse-power, greatly increased strength and rigidity of all parts, very precise balancing and the incorporation of a flywheel.

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CZECHOSLOVAKIA

THE FUTURE OF THE CZECH MOTOR CAR INDUSTRY.

By J. KOSTAL. (From *Strojnicky Obzor*, Vol. 26, No. 11-12, 1946, pp. 101-102.)

Considerable attention is being paid in America and in Great Britain to the development of automobiles in European countries. This interest is natural, being an integral part of the study of export markets. Lawrence Pomeroy, technical editor of The Motor, London, recently addressed a meeting of the Society of Automotive Engineers in French Lick, Indiana, U.S.A., on "European Automobiles" (See Mechanical Engineering, July, 1946).

In this connection it is considered worth while giving the opinion of a leading Czech authority on the desirable trend of design in his country. He advocates the two-stroke engine, and indeed before the war there were some successful types such as the Yava, Zero, and the "Z" of the Bren gun firm known beyond Czechoslovakia (cf. THE ENGINEERS' DIGEST, Vol. 2, No. 7, July, 1941. "The Development of Two-Stroke Engines for Motor Cars," by H. J. Venediger) Prof. Kostal considers fuel injection as desirable and practicable. In this connection it may be of interest to re-read the paper of P. M. Heldt on fuel injection for aircraft engines in the September, 1941 issue of "Automotive Industries," where the rather uneconomical nature of fuel injection is stressed

THE war years interrupted the normal activities of Czech motor car firms, and the pre-war types after the long interval, appear to be somewhat too obsolete to warrant a resumption of their production. Czech automobile manufacturers will, therefore, have to enter the market with new types.

It appears improbable that Czech designs will follow the same lines as, for instance, American, which have in the past shown the way, because economic considerations and the price of fuel will determine different and independent trends.

Apart from conditions resulting from a development of domestic sources of fuel and from the general problems of automotive design, the question of two- or four-stroke motors is of interest.

For several years prior to the war a number of Czech firms were building two-stroke motors, and these have been more generally accepted in their country than abroad. Originally, these motors were intended for the inexpensive car, but later found acceptance in the medium type of car. These motors were cheap and reliable, but in the types of car used, were less economical than four-stroke motors. However, information at present available indicates the possibility of producing a cheap yet economical two-stroke motor in great quantities. This the Czech industry will soon have to consider. Fuel injection into the cylinder heads of aircraft engines made great progress during the war and the application of this principle to two-stroke motor-car engines appears to be very promising since it would improve efficiency. The filling of the cylinder with air and the subsequent fuel injection is designed to put an end to the loss of fuel otherwise present in two-stroke motors, and thus make this type of motor more economical.

It is assumed that the fuel pump required is feasible of solution on the basis of sufficient reliability and cheapness. These are the problems, and Czech planners will first have to decide the question of two- or four-stroke motors, before enlarging upon further issues connected with the motor car industry of a country that is to produce cars within a planned economy.

FRANCE

THE CALIBRATION OF VENTURI METERS.

By A. VIBERT. (From *Le Génie Civil*, Vol. 123, No. 11, June 1, 1946, pp. 138-141, 2 illustrations.)

DESPITE the universal application of venturi counters, the accuracy of their readings is still questioned by some engineers. This may be due to irregularities in performance which have occasionally been noticed. Industrial venturi meters consist essentially of :

(1) A venturi tube proper which produces a pressure drop in the fluid flow to be gauged. This drop results from a progressive narrowing of the flow cross section. The measurement of the physical phenomena thus produced allow of an estimation of the volume passed through the venturi.

(2) An indicating instrument, which indicates and records the instantaneous flow and, by a special device, gives the total volume passed during time T.

This latter instrument being a precision mechanism is subject to disturbances inherent in devices of this type. The performance depends on the design, the manufacture and the maintenance in service.

The tube itself gives constant results and complies well with its calibration curve provided it is installed under hydraulically satisfactory conditions, apart from the effects of roughness of the internal walls which depend on the nature of the fluid.

In general, the user does not distinguish between the two parts which make up a venturi meter and a fault of the instrument may be blamed on the tube and *vice versa*.

Doubts about the tube are due to the fact that the theory is based on the theory of perfect fluids which is very well known ; but this cannot be said about viscous fluids. The venturi tube shows a very constant calibration coefficient, yet many engineers regard its indications as valid only within narrow limits.

We propose in this article to establish a theoretical method to calibrate these instruments *a priori* by an approximation at least equal to that obtainable by the more or less empirical means which have been used so far.

The calibration coefficient.—Let us consider a venturi tube the divergent portion of which is characterized by the inlet diameter D, the throat diameter d, and the length L. The general equation of motion between these two sections is then given by :

$$\alpha \frac{U_o^2}{2g} + \beta \frac{P_o}{\pi} + Z_o = \alpha' \frac{U_c^2}{2g} + \beta' \frac{P_c}{\pi} + Z_c + \xi$$

where U_o, P_o, Z_o are the average speed, pressure and height at the entering cross section ω_o ; and U_c, P_c, Z_c the same quantities at the throat ω_c.

α and α' are the classical coefficients taking into account, for each cross-section under consideration, that instead of using the energetic averages of the speed given by

$$U_{eff}^2 = \frac{\int_\omega V^3 \, d\omega}{\int_\omega V \, d\omega}$$

the averages are defined in the ordinary way by

$$U = \frac{1}{\omega}\int_{\omega} V d\omega$$

Similarly, β and β' take into account that P, the pressure measured, may be different from the effective pressure.

$$P_{eff} = \frac{\int_{\omega} PV d\omega}{\int_{\omega} V d\omega}$$

Finally ξ represents the losses between the two cross-sections.

Not much is known about the values of α, α', β, and β' which, in the convergent part of the venturi seem to be close to unity. This is generally accepted and in agreement with the theory given below.

Neglecting the vena contracta (which is permissible in a well designed venturi tube) the equation of motion changes into[*]:

$$U_c = K A \sqrt{2 g H}$$

where the following definitions and abbreviations are used :

$$K^2 = (H - \xi)/H \; ; \; H = (P_o - P_c)/\omega + Z_o - Z_c \; ;$$
$$A = \sqrt{m^2/(1 - m^2)} \; ; \; m = \omega_c/\omega_o.$$

A depends on the geometry of the venturi.

K is called the coefficient of the venturi tube. It depends essentially on the losses in the converging portion and any method allowing to calculate these losses *a priori* would contribute to the solution of the problem of calibration.

The factors determining these losses are : (a) the roughness of the inner walls ; (b) Reynolds number and ; (c) the shape of the tube, since the convergence of the stream lines causes an increase in turbulence.

From practical experience the following two facts have been established :

(1) The coefficient of calibration K depends on Reynolds number characterizing the flow in the venturi.

(2) With equal roughness K decreases with the diameter. Before going into the details of these two aspects connected with the same question we shall determine, by simple means, the extreme limits of the problem.

Order of magnitude of the absolute error in the determination of K.—For a divergent cone given by its length L, its diameters D and d, and the discharge Q, the total losses will be between a lower limit ξ_{min}, determined by an equal discharge Q, equal length L but constant diameter D, and an upper limit ξ_{max} which is, under the same conditions, the loss in a tube of diameter d.

Assume that

$D = 1{\cdot}00$m (40 ins. approx.),
$d = {\cdot}50$m (20 ins. approx.),
$L = 2{\cdot}00$m (80 ins. approx.)

For a discharge $Q = 785$ 1/sec (48000 cu. in./sec), corresponding to an average speed $U_o = 1$ m/sec (3 ft/sec) at the entrance, we have :

$$H - \xi = \frac{U_o^2}{2g}\left(\frac{1-m^2}{m^2}\right) = {\cdot}765\text{m (30 in.)}$$

For the same discharge, the total loss in a new cast iron pipe of 1m (40 in.) diameter and 2m (80 in.) length we have according to Darcy

$$\xi_{min} = {\cdot}00208$$
and for $D = {\cdot}50$m (20 in.)
$$\xi_{max} = {\cdot}06816.$$

Accordingly we find

$${\cdot}955 = K_{min} \leqslant K \leqslant K_{max} = {\cdot}9984.$$

For a geometrically similar venturi tube with $D = {\cdot}05$, $d = {\cdot}025$ and $L = {\cdot}01$ one obtains : $0{\cdot}94 < K < 1$ (approx.).

This shows that the maximum error which can result from the calibration is of the order of 4 or 5 per cent and consequently appreciably less than is sometimes thought possible.

Verification of K as a function of the discharge.—In the laminar range the losses are simply proportional to the average speed, in the case of high turbulence they are proportional to the square of the average speed. Between the two ranges the losses vary in proportion to a power (between 1·75 and 2) of the average speed.

For venturi tubes the critical Reynolds number is found to be approximately 300,000. K is constant for $R > 300{,}000$ and decreases for values of $R < 300{,}000$.

Equivalent conduit.—Due to the convergent portion of the venturi tube the losses vary from point to point along a stream line. To this difficulty is added another which arises from the fact that the Reynolds number also varies along a stream line ; e.g. water at 15 deg. C (59 deg. F) has a dynamic viscosity of 1138 centipoises and consequently the Reynolds numbers are :

$$R_D = 880{,}000\, U\, D$$

at the entrance and

$$R_c = 880{,}000\, U_c\, d = 880{,}000\,(UD)/\sqrt{m}$$

at the throat.

But it seems that these obstacles can easily be overcome if the convergent portion is replaced by an "equivalent pipe" of a diameter between D and d giving the same losses. The equivalent diameter can be calculated as follows :—We consider the convergent portion to be made up of short lengths dx, the diameter at any distance x from the throat being given by D_x. For equal loss the diameter δ of the equivalent pipe is then given by :

$$\frac{L}{\delta^q} = \int_0^L \frac{dx}{D_x^q}$$

where q depends on the formula adopted to express the loss. Generally :

$$\mathcal{J} = \lambda \,.\, (U^2/2gD)$$

with $U = Q/\omega = (4Q)/\pi D^2$ one obtains $q = 5$

For a cone

$$D_x = d + \frac{(D-d)}{L}\, x$$

which yields $\delta = D \sqrt[5]{\dfrac{4n^4(1-n)}{1-n^4}}$ where $\dfrac{d}{D} = n = \sqrt{m}$

The Reynolds number corresponding to this diameter is an average of the Reynolds numbers relative to the different sections calculated from :

$$R_m = \frac{1}{L}\int R x\, dx$$

These relations enable us to predetermine K for the two regions $R > 300{,}000$ and $R < 300{,}000$

Reynolds number $> 300{,}000$. *Variation of K with diameter.*—In this case the losses are proportional to the square of the average speed. These losses consist of two parts,

(1) outer and inner friction (viscosity),
(2) turbulence ;

in the case of a convergent conduit there is also a third part due to the "super-turbulence" caused by the inclination of the inner walls towards the axis. This inclination has effects similar to roughness.

The simple statement that for Reynolds numbers $> 300{,}000$ K is strictly constant, shows that the losses in the convergent part are proportional to the square of the speed, whatever the finish of the interior surface. On the other hand, Stanton and Panell have given evidence that for smooth cylindrical tubes the losses cannot be proportional to the square of the speed,

whatever the value of the Reynolds number, but vary with the power 7/4 according to Hagen's law. Consequently, it is the super-turbulence produced by the angle of the inner walls which causes the proportionality between the losses and the square of the average speed ; this justifies the comparison with the effects of roughness.

If this is true it should be possible to calculate K on the basis of Nikuradse's work. This we have done and the result has surpassed by far anything we could have hoped for.

Nikuradse found that

$$\frac{1}{\sqrt{\lambda}} = 1{\cdot}74 + 2 \log \frac{r}{\epsilon}$$

ϵ being the absolute, and ϵ/r the relative roughness.

He has also shown, however, that this value of λ which is independent of Reynolds number, holds good only if the latter is beyond a critical value. This is supporting evidence for the equivalence of inclination and roughness. Consequently each angle of inclination should correspond to a definite absolute roughness. This we have verified.

Investigating this correlation, experiments have been made with convergent tubes of the ordinary industrial type (cast iron, steel, or reinforced concrete),

TABLE I.—Values of the coefficient K for different diameters and for Reynolds numbers > 300,000.

D	d	δ	K (theoretical)	K (experimental)	$\Delta = K_{th} - K_{exp}$ in per cent
0·05	0·025	0·033	0·977	0·975	+ 0·2
0·10	0·05	0·066	0·984	0·980	+ 0·4
0·20	0·10	0·132	0·986		
0·50	0·25	0·330	0·988	0·985	+ 0·3
1·00	0·50	0·660	0·990	0·990	0·0
1·50	0·75	0·990	0·992		
2·00	1·00	1·320	0·993		
3·00	1·50	1·980	0·994		

with the standard angle of 21 deg., $m = n^2 = \cdot25$ and $\cdot05 \leqslant D \leqslant 3{\cdot}00$m ($2 < D < 120$ in.). For this type a fictitious roughness $\epsilon = \cdot00125$ led to the results summarized in Table I from which can be seen a remarkable agreement between theory and experiment.

Reynolds number < 300,000.—In this case, we require another expression for λ which is proportional to the reciprocal of R. It is therefore tempting to compare the role of the superturbulence with an undulating roughness. This gives for λ values situated on parallel straight lines either on the essentially straight part of von Karman's curves or on the straight lines of Blasius.

The latter curve is given by the equation :

$$\log \lambda = \log \cdot3164 - \cdot25 \log R$$

In our case we assume a similar equation for λ, namely

$$\log \lambda = \log A - \cdot25 \log R$$

solutions of which pass for $R = 300{,}000$ through the point previously determined. Table II gives the results for $R = 4000$ where deviations from the experimental values are greatest.

TABLE II.—Values of the coefficient K for different diameters and for a Reynolds number $R_e = 4000$.

D	d	δ	K (theoretical)	K (experimental)	$\Delta = K_{th} - K_{exp}$ in per cent
0·05	0·025	0·033	0·9415	0·948	—0·65
0·10	0·05	0·066	0·9521	0·956	—0·39
0·20	0·10	0·132	0·9627		
0·50	0·25	0·330	0·9717	0·964	+0·7
1·00	0·50	0·660	0·9758	0·969	+0·6
1·50	0·75	0·990	0·9787	0·976	+0·2
2·00	1·00	1·320	0·9807		
3·00	1·50	1·980	0·9820		

The experimental and theoretical values of K agree very well and the differences given are smaller than the errors incurred when determining K experimentally.

RUSSIA

NEW METHODS OF MELTING IN THE CUPOLA.

By V. M. ANDREYEFF. (From 3rd Technical-Scientific Conference of the Kalinin Polytechnic Institute, Leningrad, September, 1944, pp. 73-75.)

LATTERLY, among high-quality cast irons, modified cast iron has reached wide application. This modified cast iron, similarly to other high-grade cast irons, can only be obtained if the liquid metal is superheated. This can be done in the cupola by several methods. Combustion can be improved and with it the temperature of the metal in the cupola raised, (A) by stimulating the process of combustion of the fuel to CO_2 in place of CO, and (B) by burning any CO formed.

The following methods of stimulating CO_2 formation seem to be worth considering :

(a) introduction of catalysts into the combustion zone ;
(b) increase of the surface of the burning fuel ;
(c) increase in the oxygen content of the blast.

So far none of these three methods has found application, but method (B), combustion of the CO formed, has been applied.

Of various attempts towards achieving this end, the best results have so far been obtained by adding a few more rows of tuyeres, so as to improve the distribution of the air supply. Properly spaced tuyeres, of suitable section, enable the air to be distributed evenly over the entire section of the cupola and to burn part of the CO formed at the main level of tuyeres. Tests have shown that by this means the coke consumption could be reduced by 10 to 20 per cent., with a simultaneous increase in output of between 25 and 30 per cent., and rise in the temperature of the metal from 20 to 30 deg. C.

The utilization of the exit gases also deserves consideration. The total heat produced by the combustion of the fuel is used up as follows : for melting and overheating the metal, about 50 per cent. ; for radiation and convection losses, about 10 per cent. ; and heat taken out to the atmosphere with the escaping gases, 40 per cent. This shows that it ought to be possible to raise the cupola efficiency by reducing the heat lost in the exit gases. Between 65 and 67 per cent. of the heat potential contained in the escaping gases is in the form of carbon monoxide. This can be used up by burning it to carbon dioxide, and thereby preheating the air supply.

A recuperator has been developed at the Institute "Zniitmash", based on the design given by Griffin, and using a pin type of cast iron recuperator. A series of tests has shown that the temperature at the spout of the cupola was increased by 50-70 deg. C. as compared with operation on cold blast. The mean temperature of the cast iron was 30 deg. C. higher when using 9 per cent coke and hot blast, as compared with the mean temperature of cast iron obtained with 15 per cent. coke and cold blast. The temperature of the preheated air reached 270-300 deg. C. At this temperature the pressure in the air chamber was between 430 and 380 mm. water column.

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Water		Crude	Treated	Crude	Treated	Crude	Treated
Cations							
Calcium	Ca	3.2	-	9.4	-	10.7	-
Magnesium	Mg	0.8	-	0.36	-	1.09	-
Sodium	Na	0.46	0.23	1.0	0.31	1.66	0.44
Total		4.46	0.23	10.76	0.31	13.45	0.44
Anions							
Carbonate	CO_3	4.2	0.24	12.4	0.29	10.5	0.57
Chloride	Cl	1.8	0.06	2.5	0.12	2.84	0.30
Sulphate	SO_4	1.35	-	3.48	0.03	11.95	-
Nitrate	NO_3	-	-	-	-	1.15	-
Total		7.35	0.30	18.38	0.44	26.44	0.87
Total ions in solution		11.81	0.53	29.14	0.75	39.89	1.31
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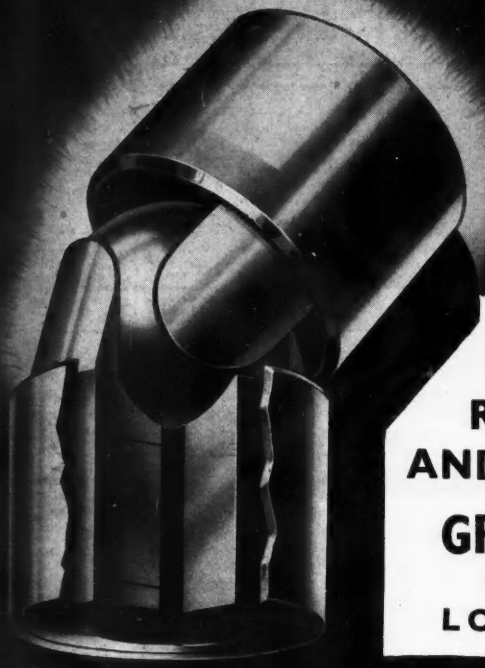

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CLASSIFIED ABSTRACTS

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AERODYNAMICS

Dynamic Similarity of Flow of Air and Superheated Steam.

By V. A. KIRSANOV. (From *Izvestia Akademii Nauk, Russia*, No. 2, 1946, pp. 227-258, 12 illustrations.)

A considerable amount of research was carried out by the author to determine to what extent it is possible to substitute air for superheated steam for model experiments on turbine blading at high, but not supersonic velocities.

The theoretical conditions for complete dynamic similarity of an air-driven model to a steam turbine are too complicated to be practicable. Apart from geometrical similarity, identity of equations of motion and initial and boundary conditions, it is also necessary to have the following characteristics identical :— (*a*) Mach number, (*b*) Reynolds number, (*c*) Prandtl number, and (*d*) adiabatic constant. And these in turn represent some ten different characteristics, all of which, to get in the right proportion to each other, is quite out of the question.

The practical possibilities are limited to obtaining geometrical similarity, identity of Reynolds and Mach numbers and identity of initial and boundary conditions. Ackeret, Keller, Sorensen and other scientists working in Escher-Wyss laboratories have proved theoretically that, provided the Mach number does not exceed 0·5 to 0·7 there is reasonable similarity between air and superheated steam. The object of the present series of experiments using normal turbine profiles, was to determine the degree of error introduced by the use of air. A full description of the experimental installation and the methods of computation are given, with a large number of comparative results, leading to the following conclusions :—

1. In the main, continuous flow, the similarity is complete.
2. In the regions of local turbulence there is a rough similarity of pheonomena but some difference in detail.
3. These local dissimilarities lead to slightly more favourable results with air.
4. With Mach numbers below 0·2 — 0·3 the differences may be said to be smaller than experimental errors.
5. Beyond that, air experiments give good comparative results but cannot be relied on for absolute figures.
6. Direction of main flow is identical for air and steam and the mean effective angle is unaffected by changes in the Mach number.
7. These experiments do not apply at all to wet steam, which would require a separate study, nor do they cover supersonic speeds. But from results obtained with a Mach number of 0·9 (where shock waves begin to be felt), it appears to be likely that the conclusions reached may hold good in the supersonic regions.

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ELECTRIC HEATING

Electrode Water Heaters.

By R. MALGAT. (From *Électricité, France*, Vol. 30, No. 118, July/August, and No. 119, September, 1946, pp. 153-159 and 178-184, 24 illustrations.)

IN electrode water heaters, an electric current passes through water between two electrodes, the water acting as a resistance. Transformation of electric energy into water heat is therefore direct and transfer efficiencies, including heat losses through the walls, of 95 to 98 per cent can be guaranteed. The heaters can be attached to any alternating current power mains of 200 to 30,000 volts and are built for power consumptions of from 15 up to 12,000 kW. producing roughly 2·8 lbs. of steam at 15 to 250 lbs. per sq. in. per kWh. Direct current cannot be used because of its strong electrolytic effect. Power input and steam or hot water production can easily be controlled automatically or by hand, by variation of cross-section and/or length, and therefore resistance, of the water path between electrodes. Sketches show principles of different control systems and electrode arrangements.

Generally, the heaters consist of a lined vessel, of an electrode block with electric mains connections and a tube arrangement for power control, and of the control devices. The electrode block is often attached to the vessel cover and easily accessible for inspection and infrequent cleaning. The heater size is approximately one fifth to one third that of a corresponding boiler, e.g. a vessel for 100 kW., 280 lbs. steam at 120 lbs. per sq. in. occupies a space 18 × 18 × 72 in. high. No storage space for coal, etc., is required nor are, for reasons of easy and reliable electrical control arrangements, the safety precautions as stringent as for boilers. No service personnel is required for wholly automatic heaters. Heaters start up very quickly supplying steam or hot water of desired temperature within several minutes. They can be used as heat or energy storage, being heated up when electric energy is cheaply available and supplying heat during peak periods. Heat losses can be kept very small and efficiency does not suffer by intermittent service. The cleanliness, ease of service and control, small size and independence of ancillary apparatus enable the heaters to be arranged near the place of usage, often in existing buildings without structural alterations. Coal is saved and atmospheric pollution avoided. Principal users of electrode heaters are chemical, textile, food, tobacco industries, plastic plants, laundries, hospitals, baths, test laboratories and central heating installations.

A detailed survey of design principles and of various types of heaters and control arrangements is supported by many schematic and photographic illustrations.

ELECTRONICS

Automatic Electronic Control.

By A. A. BULGAKOV. (From *Vestnik Inzhenerov i Tehnikov, Russia*, No. 3, 1946, pp. 104-110, 17 illustrations.)

SEVERAL electronic control schemes for heavy machinery were detailed and tested by the Russian Electrotechnical Institute in 1939-41 but owing to the war they were not taken into operation. The article under review gives their description in brief, diagrammatic arrangements, speed and voltage characteristics, frequency characteris-

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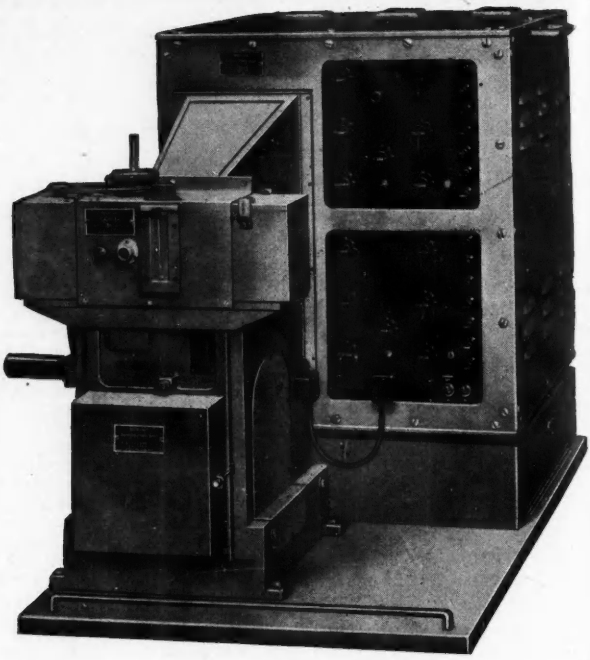
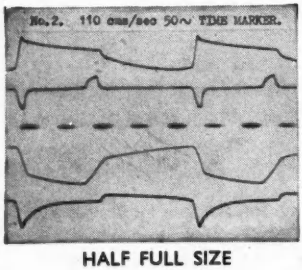


This equipment is the latest development in the use of sealed-off Cathode Ray Tubes in Multiple Recording technique as pioneered by Cossor. The robust design is intended to make it suitable for field tests, including Traction Recording, Marine and Motor testing and other industrial investigations. The equipment comprises a Model 440 Twin Tube Unit using two 4½" dia. Double Beam Tubes, a Model 423 Film Camera Unit and a Model 430 Drive Unit. It provides simultaneously four independent records plus one central electro-optical time marking on 70 mm. unperforated film or paper. By means of interchangeable gears and/or motors a film speed range of 0.005-250 cms/sec. is obtainable with a tube spot writing speed up to 2 kms/sec. Two or three units may be mounted one above the other and driven by the single Drive Unit to provide up to a maximum of twelve simultaneous records plus three time/event markings.



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tics of various parts of the systems, some oscillograph records and a brief description of methods of calculation. The schemes described are :—

1. Electronic control of voltage for a mercury rectifier feeding a multiple sychronised drive of a paper making plant, requiring very stable voltage conditions at variable loads and mains voltages.
2. Electronic speed regulator for a 350 hp. rolling mill requiring speeds varying from 40 to 800 r.p.m. to be controlled to within 1 per cent.
3. An alternative design of regulator.

The methods of calculation are orthodox but the rest of the information appears to be the result of work on independent lines and as such may be of interest, even though somewhat "dated," to specialists in this country.

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INTERNAL COMBUSTION ENGINES

Water Injection in a Spark-Ignition Engine.

By W. P. GREEN AND C. A. SHREEVE. (From *Transactions of the A.S.M.E.*, Vol. 68, No. 5, July, 1946, pp. 541-545, 11 illustrations.)

THE present war witnessed a revival of interest in the use of water as a coolant and detonation suppressor inside the engine cylinder. Aircraft engines, when operated with water injection, have been boosted in output above normal sea-level capacity and have functioned satisfactorily. This paper gives a review of past tests on the use of water as a coolant and detonation suppressor in petrol engines. It presents new data regarding the use of water in very high compression ratio engines to promote increased part-load efficiencies with low octane fuels.

IRON AND STEEL

Catalytic Agents in Blast Furnaces.

By M. I. KOROBOVA AND N. I. KOROBOV. (From *Izvestia Akademii Nauk, Russia*, No. 4, 1946, pp. 567-577.)

IN the search for greater speed and efficiency of the reducing process which takes place in a blast furnace, a great advance has been made by various methods of preparing the ore, such as crushing, sorting, roasting and briquetting.

As regards the physical conditions in the furnace, further improvement can hardly be expected as far as temperature is concerned ; but it is probable that increased pressures will be used. This would necessitate a complete re-design of the furnaces and is therefore a matter of long-term development.

For more immediate improvement, without drastic changes of the process, the use of catalytic agents is now receiving a great deal of attention.

Various metals, akin to iron, such as Ni, Co, V. are effective, but from the economic point of view their use on a large scale is unlikely. Experiments have recently been carried out in Russia with such oxides as K_2O, Na_2O and CaO and it has been found that the use of 5 to 8 per cent of Na_2O will speed up the reducing process 2 to 10 times. This was tried out on a commercial scale at the Chusovsky works using natural rocks containing Na_2O in connection with a titanium-magnetite ore with good results.

The best known and most promising catalytic agents, however, are solid carbon (in its most active form of soot) and hydrogen, the latter being effective both in its uncombined state and in the form of steam. Fear of the cooling effect of the " cracking " of steam has encouraged experiments with uncombined hydrogen, but this fear is unfounded, because the loss of heat can be fully compensated by increased temperature of blast. The use of steam has great advantages, on account of costs and because it has better catalytic properties than pure hydrogen.

It is surprising, therefore, that while the addition of water or steam to the blast has been practised for some time as a means of obtaining an equally distributed moisture content in all weathers, and while the addition of pure hydrogen as a catalyst has also been tried, the use of steam as a catalyst seems to have been overlooked ; on the contrary much trouble has been taken sometimes to devise means for drying the blast.

A scheme for the use of steam to expedite the reducing process has now been worked out by L. Y. Gabrielian and tested at full scale by the Magnitogorsk and Kuznetsk steel combines.

It is too early to speak of this process as more than experimental, and to imagine in what form it will finally be used ; but it is probable that the amount of steam will be about 6 per cent and the blast temperature 1000 deg. C. The increase of temperature of the blast required is theoretically 74 deg. C. per 1 per cent of steam. So far, however, not more than 800 deg. C. and 4 per cent steam have been used in practice.
The following advantages were found to result :—

1. Speeding up of the process by catalytic action.
2. Bigger and more uniform oxidising zone, with less clogging and better slag formation. This, again, results in speeding up the process.
3. Reduction of the amount of coke, due partly to the fact that hydrogen compensates for carbon but mainly to the smoother operation and quicker turnover.
4. Easier control of temperature conditions. Overheating is avoided, and with it various forms of clogging, increased resistance to blast, excessive silicon content in the pig, etc.

During a 21-day trial under far from ideal conditions of steam control, an almost 30 per cent improvement of output could be recorded with an average blast temperature of 780 deg. C. and a steam content of 1·5 to 5 per cent, say an average of 3 or 3·5 per cent.

MACHINE SHOP PRACTICE

Improved Method for Making Fine Diamond Dies.

(From *Materials and Methods, U.S.A.*, Vol. 24, No. 3, September, 1946, p. 679.)

A METHOD for making fine diamond dies, 0·0004 to 0·015 in. in size, in roughly one-third the time required by previous European and American methods, has been developed by The Bureau of Standards, Dept. of Commerce.

The European and American die makers customarily used a star drilling action in making diamond dies. This called for a drilling machine with a horizontal, single spindle making 3,500 r.p.m. The machine carried a sewing needle for the drill. The diamond, mounted on a second spindle, oscillated against the sharpened end of the drill. Frequent regrinding of the needle was necessary. The method was tedious and took from 115 to 170 hours to produce a die of 0·001 in. or less.

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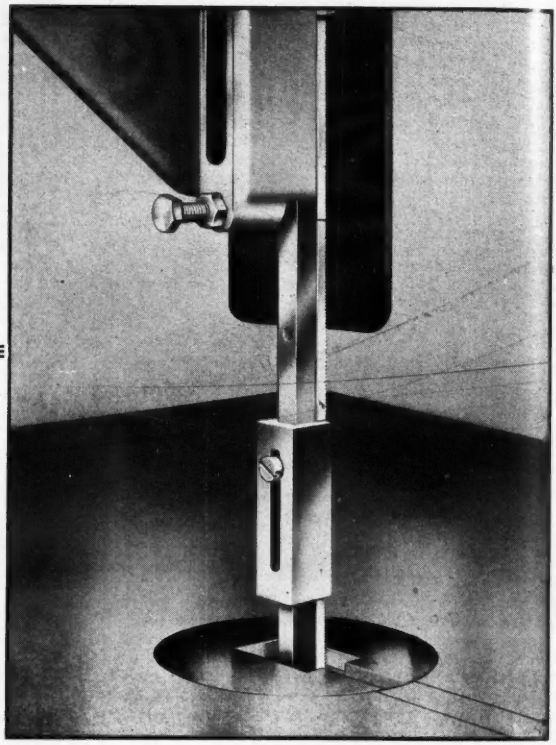

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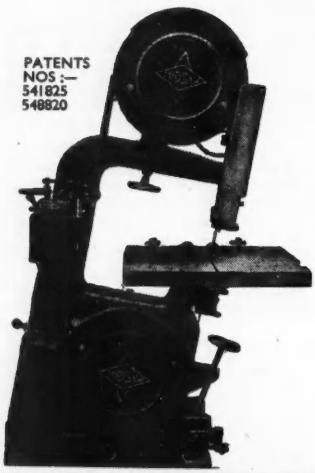

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Machine Shop Practice—*continued.*

Use of electric current in the drilling process is the chief innovation developed at the Bureau of Standards. The equipment consists of a variac, rheostat transformer, alternating current ammeter, quenched spark gap, and a support for the diamond and electric drill.

To form the primary cone, a pilot hole is foredrilled through the diamond by means of an electric spark. Then, by working through the pilot hole with a light commercial drill, the contour of the cone can be reamed or shaped without dulling the drill. The operation takes about 3 hours for one die.

Drilling of the secondary cone, also done electrically, takes 1 hour. Thirty hours is considered necessary for finishing and polishing the secondary cone and bearing. Previous methods have not given sufficient emphasis to this phase of the work. However, polishing is done with a multiple head drilling machine, operating 24 hours a day and requiring scant attention, so that the cost per die is small.

Altogether, with the methods used by the Bureau of Standards, a die can be made in about 40 hours.

Another objection to the foreign die-makers' practice is that of bruting-cutting one diamond with another to form parallel surfaces. This practice results in strain, flaws, or fractures in the die.

MANAGEMENT ENGINEERING

Foundry Organization and Management.

By K. G. Wiss. (From *Gjuteriet, Sweden*, Vol. 36, No. 8, August, 1946, pp. 127-137, 4 illustrations.)

During the last decades much work has been devoted to the technical advancement of the foundry industry. For various reasons, however, organization and management problems have not received as much attention as the technical questions. With this in mind, the Foundry Organization and Management Committee of the Mekanförbundet has worked out organization sheets for gray iron foundries with 25, 75, 125, and 350 workers respectively, designed to serve as a general organization pattern for such foundries. The duties, desired background, and knowledge of superintendents, metallurgists, foremen, etc., are briefly outlined. Since the Swedish foundry industry is specialized only to a very small degree, it is believed that these recommendations will prove quite useful.

MATERIALS TESTING

Improving Machine Parts Through Strain Measurement.

By R. G. Anderson. (From *Machine Design, U.S.A.*, Vol. 18, No. 9, September, 1946, pp. 137-142, 9 illustrations.)

Practical determination of surface stresses by strain measurement can furnish the designer with much valuable information leading to improved life of highly stressed parts. In this article the author is dealing with special cases and interprets the results of some actual strain measurements on machine parts.

He classifies the major stresses in an operating machine and their origins as follows:—

1. *Imposed stresses resulting from operating loads.*

They can be calculated by the use of *Stresscoat*, and if bi-axial strains must be measured, two gauges can be applied at each interesting location, one normal and one parallel to the direction of the principal strain. Formulas are given for computing the indicated stress resulting from bi-axial strain; but it should not be overlooked that a high degree of accuracy can be obtained only if the principal strain is within the elastic strain of the material and if this strain results from pure tension or compression.

2. *Residual stresses arising on the release from non-uniform plastic deformation.*

They may be set up in a part during casting, fabrication, heat treatment machining or assembly. The quantitative determination of these stresses is generally destructive to the part, but a qualitative investigation may be made by means of a special method of using *Stresscoat*.

3. *Pre-stresses introduced during assembly.*

An increase in the maximum allowable operating speed of rotating parts can often be obtained by so-called "overspeeding" and results in pre-stresses. Another example of this kind of stress is screw thread fastening during assembly, whereby an improvement in life of the parts concerned can be obtained.

4. *Thermal stresses set up during operation as a result of thermal gradients*, in other words: these stresses are induced by restraining the normal expansion or contraction, which would accompany a change in temperature.

It is obvious that combinations of these various stresses may occur in a special case and the determination of the resulting stress may then be extremely difficult and may need careful analysis. Even then the possibilities of designing a machine member free of stress concentration are extremely limited.

As regrettable as it may appear from a scientific point of view, the investigation of the problem dealt with in this article gives no definite answer to the designer, how to form the machine part he is just developing, and which dimensions he should choose; but valuable hints can be obtained about the nature of the most suitable material and its best distribution within the part. For economical reasons designs should be proportioned to take advantage of any pre-stress, which can be beneficial, and although direct measurement of strain requires considerable research and study, it has proven its worth in strengthening many machine parts without increasing their weight.

Notes on the Generation of Synchronous and Superimposed Alternating-Bending and Torsional Stresses.

By O. Puchner. (From *Schweizer Archiv*, Vol. 12, No. 9, September, 1946, pp. 289-293, 6 illustrations.)

So far fatigue-strength and durability of materials and constructional parts have been determined only by means of testing machines of a type especially designed and constructed for this purpose. This article, however, shows the possibility of carrying out such investigations in the usual standard machines.

The ratio of bending strain to torsional strain required in a special case is obtained on torsion machines and bend testing machines by chucking the specimen to be tested in such a way that the axes of the machine and of the specimen form a certain angle to each other. If the two axes coincide or are at an angle of 90 degrees to each other, the specimen is only subjected to either torsion or bending.

For intermediate positions, the deformations of the specimen which are forced on to the specimen by the drive of the machine, must be taken into consideration. They are the result of the relations between the bending moment and the twisting moment, and they depend on the shape of the specimen and its cross section as well as on the modulus of elasticity and the modulus of shearing of the material to be tested. Beyond it an additional moment originates working vertically to the longitudinal axis of the machine in the case of bending machines, and in the plane of the longitudinal axis in the case of torsion machines; obviously an additional strain of the various machine parts results from this last moment.

SICK WORKERS GO PENNILESS

Cold charity was the lot of the sick worker in mid-Victorian England. Many faced destitution when small savings were gone. Sickness and misery prevailed in workrooms which were often crowded and insanitary.

To-day, the worker has sick benefits, bright workrooms, canteens and music, and factory hygiene has come to stay. To help keep health standards high, many hundreds of factories have installed the Izal System of Industrial Hygiene. It aims at safeguarding the worker's health at all points where colds, 'flu and other infections are likely to develop. Wherever it is employed the sickness rate usually shows a decline.

The Izal System is simple to install and maintain in factories large or small, and seldom requires extra labour. If you have any factory hygiene problems write to Newton, Chambers & Co. Ltd., Thorncliffe, Nr. Sheffield, who will gladly arrange for an expert to survey the factory premises and make recommendations free of charge.

THE IZAL SYSTEM OF INDUSTRIAL HYGIENE

PLASTICS

Dielectric Properties vs. Temperature of Thermosetting Moulding Material Preforms at Radio Frequencies.

By T. HAZEN. (From *The Electrochemical Society, U.S.A.*, Preprint 90-13, 1946, pp. 193-204, 5 illustrations.)

IN connection with the use of high-frequency dielectric preheating in the moulding of plastics, a quantitative knowledge of the room temperature dielectric properties of the materials used is of considerable value. Of even greater value is a knowledge of how these properties vary as the temperature increases during the preheating and curing cycle. Data for a number of typical preformed thermosetting moulding plastics are given for the temperature range 24 deg. C. to 200 deg. C. (75 deg. F. to 400 deg. F.) at a frequency of 30 megacycles/sec., and the method of measurement is described. A common characteristic of these data, viz. the existence of a maximum in the dielectric loss factor *vs.* temperature curve, accounts for the success of radio-frequency dielectric preheating methods in the commercial moulding of thermosetting plastics, in contrast to the difficulties often encountered in the case of many of the thermoplastics.

RAILROAD CIVIL ENGINEERING

Railway Permanent Way Consolidation by Electrochemical Method.

By A. DRAGI. (From *Zheleznodorozhnyi Transport, Russia*, No. 5-6, 1946, pp. 75-78, 2 illustrations.)

CONSOLIDATION of unstable ground to prevent subsidence under the weight of a railway line can be achieved in many ways, depending on its composition, e.g. injections of bitumen, clay and silicate, cementation, etc. None of these methods can be considered as a complete success in the case of subsoils mainly composed of clay or river mud.

In the search for a better solution of this problem an electrochemical method has been tried with considerable success. This consists in passing a direct current through the ground to be consolidated, which results in changes of the structure of the clay, enabling it to take much heavier loads and to resist the liquefying action of water.

The recommended procedure in approaching a particular problem is to analyse the untreated ground; to determine in a laboratory the "dose" of electricity required to effect the necessary cure; to plan the operation as a whole, and then to try it out on a small experimental sector and observe the results.

Contrary to what might be supposed, the consolidation of this nature does not require heavy currents (i.e. it is not a matter of burning or baking the clay), although the total amounts of KWh. used during a number of days is fairly large. A satisfactory formula for the time t of operation recommended by Borkov is:—

$$t = K \times 360 \times \frac{PRl^2}{v^2}$$

where

p = required increase of bearing capacity in kg./cm².
R = specific resistance in ohms per metre length and per sq. mm. cross-section.
l = distance between electrodes.
v = terminal voltage.
K = a coefficient determined experimentally, which in the present instance proved to be 3×10^{-4}.

To give an idea of how these figures work out, the author was using, on his test-sector, electrodes 30 in. apart, 40 and 60 in. deep (on two different sectors each of about 50 sq. yards), and because only 8·5 KW. were available, he applied this current for 37 days to one sector (66 KWh. per cub. yard) and 31 days on the other (55 KWh. per cub. yard).

After severe frosts during the winter and torrential rains in the spring the sectors treated were practically unaffected by the weather except for a little surface erosion by rivulets, while an adjoining untreated sector showed big surface changes and considerable subsidences.

Experimental results showed that a needle of definite weight dropped into untreated ground penetrated six times as deep as in the treated sector. Another interesting test was as follows:—a sample was suspended in water in a wire cage from a graduated float so that its rate of disintegration could be observed by the rising float. Untreated samples were entirely washed away in 1 hour 43 min. Treated samples taken near the anode did not begin to lose weight until after 4 hours and those near the cathode after 8 hours.

The new method is regarded as a success and will be used this year by the South Ural Railway on three sectors of their line.

VIBRATIONS.

Notes on the permissible Ratio of Critical to Running Speed of Multi-stage Pumps.

By A. VOSKA. (From *Schweizerische Bauzeitung*, Vol. 128, No. 14, October 5th, 1946, pp. 173-177, 13 illustrations.)

WHEN designing a large pump, it is usual to make an estimate of the critical speed in the early design stages, since it is often found that the first critical speed comes to lie in the proximity of the speed at which the pump is to operate, especially in high pressure pumps with long shafts. In these cases the question arises as to the permissible ratio ϵ, i.e. the ratio between running and critical speed. Practical experience with steam turbines shows that, for running speeds below the first critical, ϵ should not exceed 0·83, whereas values of between 1·3 and 2·8 are recommended for running speeds above the first critical speed. There is reason to believe that similar values will hold good for pumps.

In this article an analytical study is made of the various factors which affect ϵ. The magnitude of the centrifugal forces which have to be taken up by the bearings is a criterion for smooth operation, and these centrifugal forces depend upon the inherent unbalance of the pump rotor. If it were possible to completely balance the rotor, ϵ could be allowed to come very close to unity. This theoretical case is examined mathematically.

The author then tries to approach conditions found in practice by assuming an inherent eccentricity, in one case constant along the whole length of the shaft $e_x = e_0$; and in the other case he assumes it to be at a minimum midway between, and zero at the bearings according to the equation $e_x = e_0 \cos \frac{\pi}{2l} x$.

After a general mathematical treatment of these two cases, two practical examples are worked out.

The author realizes, however, that an application of his mathematical treatment to practical problems is not conclusive since too many unknown factors require numerical assessment. Theory can only serve to give an indication of the relative importance of the various factors involved and to convey a general idea of the magnitude of the forces acting upon the bearings and their relation to the safe operating speed range.

As to the influence of water friction, it is found that this influence makes itself felt only within a region of $\pm$ 0·6 per cent of the critical speed, in which it tends to counteract deflections, but that it is negligible outside this region.

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The Philidas is a one-piece, all metal nut, made in steel and all the alloys normally used in industry. It can be fixed to all the usual types of screw threads. No amount of vibration can move it. No extremes of heat or cold affect it. Once fixed (whether screwed to a face or unsupported), only a spanner can move it. It can be moved and replaced innumerable times, without damaging the bolt thread.

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NEWS OF THE MONTH

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.
Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- PERSONAL
- NOTES

PERSONAL.

Edgar Allen & Co., Limited, regret to anounce the retirement, for reasons of health, of **Mr. T. H. Skelton,** Manager of their Electric Melting and Tropenas Steel Converting and Foundry Heat Treatment departments. He will be succeeded by **Mr. W. H. Everard.**

Mr. K. S. Arnold, M.I.Mech.E., general manager of Fawcett Preston and Co., Ltd., has been appointed a director of the company.

The appointment of **Sir Alwyn Crow, C.B.E., M.A., D.Sc., F.Inst.P.,** as head of the Scientific and Technical Services of the British Supply Office in Washington is announced. Sir Alwyn will be responsible for maintaining liaison with the United States authorities on all scientific matters except atomic energy, for which the Ministry of Supply is responsible.

Mr. H. Francis is retiring from the board of Kayser Ellison and Co., Ltd., Sheffield, after sixty-seven years' service with the firm.

Sir William Halcrow, on taking over the duties of President of The Institution of Civil Engineers, delivered his Presidential Address on Tuesday, 5th November, 1946. This will be the 128th Session of the Institution.

Sir William Halcrow was a member of the Panel of Engineers appointed by the Minister of Fuel & Power which reported recently on the Severn Barrage Scheme. Among other positions which he has filled are those of Consulting Engineer to Government Departments, L.P.T.B. (for tube railways) and other Public Corporations and Companies, including the North of Scotland Hydro-Electric Board; Member of War Cabinet Engineering Advisory Committee; of Advisory Council of Building and Civil Engineering Industries to Minister of Works and of Advisory Council of the D.S.I.R.

Mr. R. B. Heywood, A.M.I.Mech.E., until recently on the staff of Rolls-Royce, Ltd., is now a consultant on photo-elastic and stress problems at 37, Bass Street, Derby.

Sir Clifford Paterson, F.R.S., has been elected President of the Institution of Engineers-in-Charge in succession to **Sir Alexander Gibb.**

Mr. W. H. Sharp has been appointed assistant managing director of Thomas Smith and Sons (Rodley), Ltd. **Mr. Frederick Johnson** has been appointed a local director of the company.

Mr. H. E. Sheardown, of Messrs. Charles D. Holmes and Co., Ltd., Hull, was re-elected vice-chairman.

Mr. Allan Stevenson, C.B.E., a director of Messrs. David Rowan and Co., Ltd., Glasgow, has been re-elected chairman of the National Association of Marine Engine Builders.

Mr. Gilbert Szlumper has been appointed London representative of the North British Locomotive Company, Ltd., 82, Victoria Street, S.W.1.

Mr. Harry Towers has been appointed a director and general manager of A. Reyrolle and Co., Ltd.

The Tyne Improvement Commission announces that **Mr. R. F. Hindmarsh,** chief engineer, is to retire early next year. **Mr. A. L. Harvey** is to be his successor.

NEW LITERATURE.

Miniature Panel-Instruments.—A new publication dealing with the range of Miniature Panel Instruments made by Elliott Bros. (London) Ltd., has been issued. This publication lists the complete range of standard products in this Group of Measuring Instruments. It deals with round, square, edgewise and sector panel mounting instruments having scale lengths from 2" up to 31". Each model is available either as a Moving Coil Instrument for use on D.C., a Moving Iron Instrument for A.C. or D.C., a Moving Coil Thermal Type Instrument for high frequency circuits, and a Moving Coil Rectifier type for use on A.C. circuits.

Comprehensive details are given of drilling and mounting dimensions and terminal stems, etc., to assist customers in preparing their equipment to receive the instruments, before delivery is effected.

The Catalogue is complete with an index and should prove useful to those members of the profession whose requirements call for the purchase of apparatus of this nature.

Die Sets.—Coley Bros. (Tools) Ltd., have issued a new catalogue and data sheet for Exacta Die Sets, which are now available in a greatly extended range of sizes to cover most requirements. A large stock of interchangeable parts is maintained ready for assembly to customers' individual requirements, and despatch can usually be effected in 3 days from receipt of orders. An entirely new type of four pillar Die Sets and an interesting range of quick-action and Work Holding Clamps manufactured by the Company is also announced.

British Standards For Lifting Tackle.—*B.S. Handbook No* 4 (pp. 355, 12s. 6d.), with an Introduction by Sir William Stanier, F.R.S. General Editor, A. L. Haas, I.S.O., M.I.Mech.E., British Standards Institution, 28 Victoria Street, London, S.W.1.

Safety in lifting affects every industry. Every workshop, factory and warehouse; every ship, shipyard, dock and quay; all buildings, power stations and construction projects have of necessity to handle loads too heavy for manual manipulation. Even the casual passer-by cannot be altogether indifferent to the integrity of appliances which may imperil his safety. During the past two decades, reponsible committees of the British Standards Institution have formulated a series of standards for lifting tackle. It is not claimed that, even now, these standards are either exhaustive or complete, but their number and importance justifies the collection and issue in a single volume of those which suspend, lift and lower a load and which are in common use in a large variety of appliances, where failure is liable to result in casualty.

The new Handbook is divided into seven sections—fibre rope, wire rope, terminal attachments for wire rope, chain, terminal attachments for chain, the materials cited in the preceding standards and a final section dealing with the statutory requirements under the Factories Act applicable to the contents of the Handbook. The first five sections contain the technical provisions of 21 published standards; the two last sections are supplementary and complementary, but none the less are essential to the purpose of the Handbook, which is to make readily accessible all matters relating to such tackle of which purchasers and users should be aware.

The last section sets forth the rules, orders, registers and certificates current at the date of publication as legal obligations.

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BUSINESS NOTES.

At the Annual Meeting of the Staffordshire Iron and Steel Institute held recently, the Presidential Address was given by Mr. G. B. Morrison, F.I.M., Technical Director of Messrs. British Rolling Mills, Brymill Steel Works, Tipton.

The subject of Mr. Morrison's address was "The Historical Development of Free Cutting Steels."

In his address, which was illustrated by lantern slides showing micro-photographs of typical free-cutting steels together with slides of interesting components made from this type of steel, Mr. Morrison covered the various stages of progress from early times to the present day, mentioning among other things certain interesting developments by his firm of various specialised steels, notably "Hitenspeed" —a high impact high tensile steel capable of being machined at the same speed as the low carbon, free-cutting type, and "Scootercase," a free-cutting case hardening steel.

At Glasgow Technical Exhibition **Philidas Ltd.**, exhibited a full range of Self Locking Nuts and devices in "INDUSTRIAL" and "TURRET" patterns.

The patented Philidas Self Locking Nut, sponsored by the Air Ministry for wide application on wartime aircraft, is now available for the demands of Industry in the important field of mechanical fastenings which must be resistant to vibration, changes of temperature, oil, etc., with no impairment of efficiency.

Two working models demonstrated the retention of efficiency of the Philidas principle after repeated application, and the anti-vibration qualities of Philidas Self Locking Nuts.

E. H. Jones (Machine Tools) Ltd. An association has been formed between Messrs. E. H. Jones (Machine Tools) Ltd., of Edgware Road, The Hyde, London, N.W.9. and Messrs. C.V.A. Jigs, Moulds & Tools Ltd., of Hove and London, by which the two Companies will work together on the production and sales of the C.V.A. range of Machine Tools.

Messrs. E. H. Jones (Machine Tools) Ltd., will be responsible for the world distribution of the C.V.A. Single Spindle Automatic Machines, Gear Hobbing Machines, Toolroom Lathes, Vertical Milling and Die-sinking Machines, Die-Presses and Injection Moulding Machines.

Mr. E. Aron, Managing Director of C.V.A., has accepted a Directorship on the board of E. H. Jones (Machine Tools) Ltd., and Mr. W. Hadley, who was previously Sales and Export Manager of C.V.A. Machine Tool Division, has been appointed General Sales Manager.

Mr. T. J. Corcoran, formerly Technical Sales Representative for the Midland area of C.V.A., also joins the staff of E. H. Jones (Machine Tools) Ltd., and will specialise in the range of C.V.A. Automatic Machines.

Messrs. Kempston Electrical Co., Ltd., High Street, Kempston, Beds., announce that they have recently acquired from the Ministry of Supply the remainder of the stock of spare parts for all makes of American Control Gear, more particularly that used on Machine Tools, and shall be pleased to quote for any spares, and to carry out repairs, also arrange for importation of parts not in stock or difficult of manufacture.

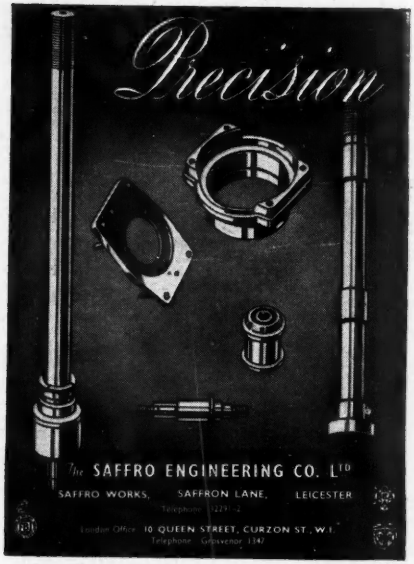
NEW EQUIPMENT.

The latest product of Messrs. C. A. Gray is a vernier gauge, plus limiting device, with a 7 ft. length of beam. This is one of several made to customers' requirements. A recent enquiry was for one with a 10 ft. beam. The gauges are marketed by E. H. Jones (Machine Tools) Ltd., The Hyde, Hendon, N.W.9.

A New Magnetic Filter.—Philips Industrial (Philips Lamps Ltd.) have produced a new type in their range of Magnetic Filters, in which the method of cleaning the filter has been radically altered to meet the special requirements of certain operations, such as grinding, where ferrous contamination is very heavy and quick and easy cleaning is of first importance.

In this new design the magnet is situated in a special chamber which seals it off from physical contact with the operating fluid in the lubricating or coolant system; the trapping cages, however, are fully magnetised whilst the magnet is in this chamber. To demagnetise the cages for cleaning purposes it is necessary only to lift the magnet from its housing by means of the ring provided. This can be accomplished without interference with any other part of the filter assembly or the circulating system. The contaminated cages which remain in the pipe-line assembly are de-magnetised as soon as the magnet is removed; thus the ferrous contamination can be quickly flushed from the filter cages by the circulating fluid.

A second outlet has been provided on this filter so that when it is being cleaned, the contamination can be drawn off into a separate draining bucket. This filter is suitable for systems with a capacity up to 700 g.p.h. Other sizes of filter will be available to deal with systems of larger capacity.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FIRST VOKES AWARD.

THE first award of the Vokes Presentation has been made to Mr. E. Carr, B.Sc., A.M.I.Mech.E., Building Engineer to the Copper Development Association, for his lecture, "The History of Copper," to members of the London Association of Engineers.

The Vokes Presentation, donated by Mr. C. G. Vokes, is to be an annual prize, awarded to members by the London Association of Engineers, for the best paper on subjects of interest to engineers.

SILVER JUBILEE SCHOLARSHIP.

THE Council of the Institute of Transport invites applications from members of the Institute for the award in 1947 of the Silver Jubilee Scholarship of value not exceeding £150. Consideration will normally be given only to applications which involve an aggregate of at least three months' absence, in periods of not less than a month each, from the candidate's employment.

The purpose of the Scholarship is to assist the selected member to meet expenses to be incurred in (*i*) travel for the purpose of studying transport, or (*ii*) specific projects of transport research, or (*iii*) full-time education at a University or other educational institution approved by the Council.

Candidates will be required to indicate the subject they desire to study, the reasons for their choice and how they would propose to utilise the Scholarship for the purpose. Candidates must have passed the Graduateship examination of the Institute but preference is likely to be given to one who has passed the Associate Membership examination and who is not over 30 years of age.

Applications must be made on a form to be obtained from the Secretary, the Institute of Transport, 15 Savoy Street, London, W.C.2, with whom it must be deposited not later than April 30th, 1947. Each application must be supported by a Member (*M.Inst.T.*) of the Institute who, by personal contact, has had the opportunity to form an opinion of the suitability of the applicant.

The member awarded the Silver Jubilee Scholarship qualifies also for the F. C. Coleman Modern Transport Award founded in 1944 by the Modern Transport Publishing Co., Ltd., in memory of the late Mr. F. C. Coleman's efforts for transport education and the prominent part he played in founding the Institute of Transport. The amount is £50 per annum for award to the Institute Silver Jubilee Scholar, to assist him in the purchase of books and equipment for the pursuit of his studies in transport.